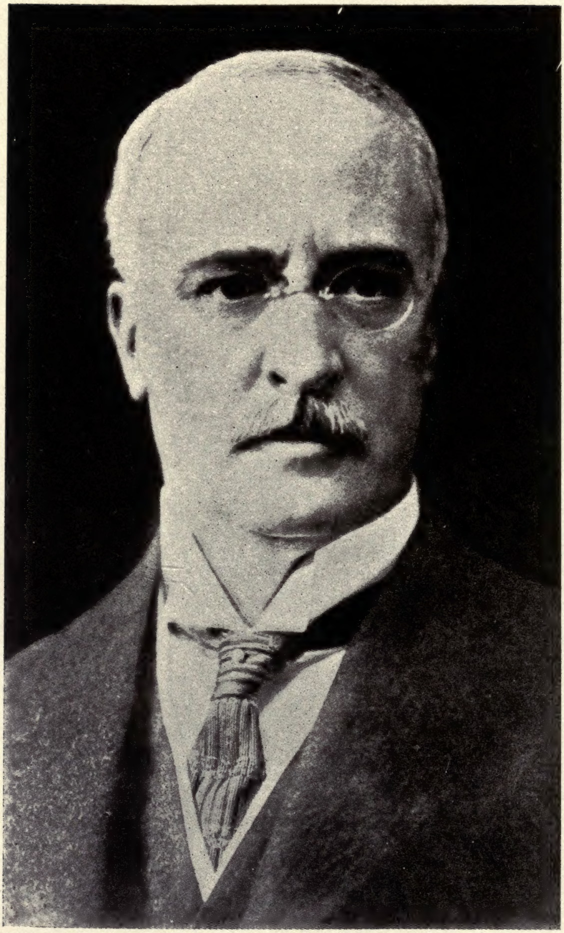
The 20th Century Guide For Diesel Operators

ROSBLOOM & HANLEY



DR. RUDOLPH DIESEL

The Inventor of the Engine operating under "Constant Pressure", named after its Originator, the Diesel Engine. Born 1858, Disappeared September 30th, 1913 while crossing the British Channel from Antwerp to Harwich, England.

The 20th CENTURY GUIDE
FOR
DIESEL OPERATORS

A PRACTICAL BOOK FOR OPERATORS, SCHOOLS, LIBRARIES AND THOSE INTERESTED IN DIESEL OPERATION

PROFUSELY ILLUSTRATED

By
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"The 20th Century Guide for Marine Engineers"
(Ramsey & Rosbloom)
"The 20th Century Guide for Automobile Operators," etc.
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FOREWORD

IN the preparation of the "20th Century Guide for Diesel Operators" all data has been carefully selected to suit the person engaged in the profession or for the use in the study of Internal Combustion Engineering.

The primary object of this valuable addition to technical publications on the subject of Internal Combustion machinery and such information as this book contains, is to instruct those interested in this prime mover in practical form.

The Authors are confident that this book will prove beneficial to those aspiring to knowledge, and with this end in view they feel that their work has not been in vain.

JANUARY, 1922.

THE AUTHORS.

WE APPRECIATE THE SERVICES AND FURNISHING OF DATA BY

Engineering Officers of the United States Navy, in particular, the late Commandant U. S. N. Captain Barthalow, Ensign G. F. De Grave, U. S. N., Internal Combustion Engineers, Operating Engineers engaged in practical operation of Diesel power, Manufacturers of Internal Combustion Machinery, etc., who so courteously assisted us in making it possible to produce the "20th Century Guide for Diesel Operators."

We also desire to acknowledge our gratitude to the Hon. Herbert Hoover, Secretary of Commerce, in so kindly encouraging the publication of the "Rules for Obtaining U. S. Licenses for Engineers on Motor-Driven Ships," and Mr. Harry C. Lord, Federal Inspector United States Inspection Service, Port of Seattle, Washington for his valuable suggestions.

The co-operation of Mr. L. B. Chapman, Professor of Naval Architecture and Marine Engineering, Lehigh University, Bethlehem, Pa., is highly appreciated.

Our thanks is also extended to the editorial staffs of the "Motorship," "Marine Engineering," "Pacific Shipping Illustrated," "Railway & Marine Review," etc., for their courtesy in extending the authors of "The 20th Century Guide for Diesel Operators" their liberal assistance.

The services of Mr. O. J. Hansen, M. E., in compilation of the subject matter, is highly appreciated.

THE AUTHORS.

WE ARE INDEBTED TO FOLLOWING FIRMS IN THE PREPARATION OF "THE 20th CENTURY GUIDE FOR DIESEL OPERATORS"

Bethlehem Steel Corporation
General Electric Company
Westinghouse Electric Manufacturing Company
Griscom-Russell Company
Dow Diesel & Pump Works
Allis-Chalmers Company
De Laval Company
Busch-Sulzer Diesel Engine Company
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The Maxim Silencer Company
Elliott Company
Cumings Oil Engine Company
Etc., Etc.

PREFACE

ONE has to marvel at the advance made in the improvements on the Diesel engine in the last ten years, while it is to be regretted that the literature on the subject, in comprehensive form, has been noticeably lacking. The task of the authors has been to link up the wonderful mechanical advance with reliable data for the engineer in the form of a complete treatise and text on Diesel Operation brought up to date.

The importance of Diesel power cannot be overestimated. Even the most skeptically inclined engineers of a few years back, as well as others interested in power generation who were tempted to treat the matter lightly have been compelled by the course of events to look upon the Diesel engine as a prime mover which is deserving of the highest consideration. The fact that its results have been of such a tremendous nature in minimizing waste, space and cost, etc., without impairing efficiency explains the trend towards universal adoption in certain lines of engineering.

The Diesel Engine is here to stay, and the progressive engineer realizing that fact finds it his duty to extend his knowledge into the Diesel field, and, as a consequence, demands more reliable and timely data on the subject. The authors have tried to answer the engineer's call by producing "The Twentieth Century Guide for Diesel Operators," in fact, they have gone farther, and tried to make this work satisfy the needs of all who are seeking enlightenment in this branch of engineering, whether they be students or experts. Neither time nor expense has been spared in an effort to make this work a success, and by success is meant "A world's standard book." The authors feel that in some ways they have accomplished much they set out to do, as there is reliable data both original and obtained which has never been within reach of the engineering world before the completion of this publication, beyond altogether the standard data expected in a text of this kind. The substance is international in character and the scope covers both land and sea operation. The writers have done their utmost to assemble in compact form all the world knows of Diesel machinery up to the present time, and having done their utmost, they now invite constructive criticism with the object of making future editions not only abreast of modern invention but also to the complete satisfaction of the engineering profession.

THE AUTHORS.

TABLE OF CONTENTS

See General Index in Back of Book.

CHAPTER I.

TECHNICAL TERMS, AS APPLIED TO DIESEL MACHINERY

Mechanical Efficiency:

The mechanical efficiency of the Diesel Engine is the net effective power developed in the engine cylinder remaining after the power is absorbed by the moving parts of the engine and by frictional resistance deducted. The mechanical efficiency is expressed as the ratio between the effective power of the engine as measured by a brake on the engine shaft. The mechanical efficiency of a Diesel engine is influenced by numerous factors, such as the type and size of the engine, the quality of the material and workmanship, the care given to details in erecting, the lubricating system, including the quantity and the quality of the lubricating oil used. If too much cooling water is used, frictional resistance, due to cylinder contraction, may be greatly increased. As the internal power and friction of an engine are nearly constant regardless of load, the mechanical efficiency decreases in the engine load.

The mechanical efficiency of engines having a two-stroke cycle is lower than that of engines having a four-stroke cycle as, in addition to the power required by the injection air compressor, there is that required by the scavenging pump.

The mechanical efficiencies of four-stroke engines at full load vary from 75 to 80 per cent, 80 per cent being usual for high grade, low-speed engines of medium and large powers. The engine efficiency, exclusive of the air compressor, is 85 to 90 per cent.

The mechanical efficiency of engines, having a two-stroke cycle, seldom exceeds 70 per cent and may be as low as 65 per cent in high speed engines. The distribution of power losses in two-stroke and four-stroke engines depends a great deal on the competency of the man in charge.

Thermal Efficiency:

The thermal efficiency of a Diesel Engine is the ratio between the equivalent in heat units of 1 horse-power and the number of heat units actually consumed by the engine in developing 1 horse-power. If based on the indicated horse-power, it is indicated horse-power efficiency, if based on the brake horse-power, it is the effective thermal efficiency.

The thermal efficiency depends chiefly on the thermodynamic cycle of the Diesel Engine, and is affected by the compression ratio (ratio between total cylinder volume and clearance volume at the end of compression) as well as the cut-off ratio (ratio between cylinder volume at time fuel valve closes and volume at inner dead center of piston or clearance volume).

The indicated thermal efficiency increases with a decrease in the cut-off ratio which contributes to the economy of the Diesel engine at fractional loads, the fuel consumption per horse-power-hour remaining nearly constant between full and three-fourths loads, and increasing only slightly at one-half load. The ignition of the fuel could be effected at lower pressures, but high compression is essential to high engine economy in Diesel Engines.

The mechanical efficiency of the engine naturally influences its fuel economy (thermal efficiency) also, but to a minor degree.

The indicated thermal efficiency of the Diesel engine having a four-stroke cycle, varies from 45 per cent at full load to 47 per cent at half load, and the effective thermal efficiency from 37 per cent at full load to 30 per cent at half load, which represents the best practice. As regards to engines having a two-stroke cycle, the figures are 10 to 15 per cent lower.

Volumetric Efficiencies:

The volumetric efficiency is the ratio between the weight of a cylinder full of air at the completion of the suction stroke and the weight of a similar volume of standard temperature and pressure. It can be determined by measuring the partial pressure of the air during the suction stroke and dividing it by the atmospheric pressure.

The construction of the engine, its piston speed, its valve gear, and temperature are factors that influence the volumetric efficiency.

The volumetric efficiency of an engine having a four-stroke-cycle differs from that of an engine having a two-stroke cycle. The volumetric efficiency influences the specific duty of the Diesel engine. During the suction stroke of the four-cycle engine, the air becomes somewhat rarefied, so that the lower the volumetric efficiency, the lower is the weight of oxygen in a cylinder full of air. The maximum quantity of fuel that can be turned by the air (oxygen) charges is also proportionately lower with lower volumetric efficiency.

The volumetric efficiency of engines having a two-stroke cycle is generally below unity, notwithstanding the fact that the cylinders are filled with slightly compressed air, as it is not possible to scavenge or remove all the gases of combustion, which vitiate the burning power of the air charge. To determine the volumetric efficiency of engines having a two-stroke cycle, it is not sufficient to know the pressure of the air that filled the cylinder (before compression begins); this value must be multiplied by the percentage of pure air present in the total weight of gas filling the cylinder.

For slow-speed four-stroke engines a volumetric efficiency of 90 per cent can be reached, which decreases to 85 per cent for high-speed engines and for extreme speeds may be lower. These values pre-suppose high-grade engines with mechanically operated valves.

Scavenging Efficiencies:

The scavenging efficiency in a two-cycle engine is the ratio between

the weight of the air contained in the cylinder at the commencement of the compression stroke and that of the mixture of air and burnt gas.

The efficiency of the charge is defined as the ratio between the amount of pure air in the cylinder at the commencement of the compression and the total cylinder volume, whilst the useful scavenging effect is the ratio between the same amount of pure air and the output of the scavenging pump.

Ratio of Expansion:

The thermal efficiency at its maximum is due to the increased temperature when the engine is at its highest production. The accomplished results in creating the full energy out of the working mediums, and the working substance.

Laws of Thermodynamics:

In the conversion of heat into mechanical energy, one unit of heat is lost for every 778 foot-pounds of energy obtained; and conversely, in the production of heat by mechanical means, one unit of heat is obtained from every 778 pounds of energy expended. It is also known, that it is impossible for a self-acting engine to convey heat from one body to another at a higher temperature without the aid of external assistance.

Specific Heat at Constant Pressure:

The specific heat at constant pressure is the amount of heat absorbed by the unit mass of gas when its temperature is raised by one degree on the thermometric scale, the pressure being kept constant, but the volume being allowed to increase.

Specific Heat at Constant Volume:

The specific heat at constant volume is the amount of heat absorbed by the unit mass of gas when its temperature is increased by one degree on the thermometric scale, its volume being kept constant, so that the addition of heat results in an increase of pressure.

Since the combustive element when heated at constant pressure necessarily expands and does work, the specific heat at constant pressure is naturally greater than that at constant volume by the amount of heat used up in doing that work; therefore the specific heat at constant volume is commonly expressed as the true specific heat. but both values require to be considered in connection with the problems of the internal combustion machinery.

Isothermal Expansion:

In isothermal expansion the temperature of the combustible element during the whole expansion remains unaltered, and hence the internal energy in the heat value remains unaltered, and the heat created is equivalent to the power in proportion to work externally.

Adiabatic Expansion.

The adiabatic expansion means that the heat is maintained on an equal base, or heat is neither gained nor lost during the expansion. The

whole of the heat being employed in doing external work, and it is evident at once that this can hardly be realized in practice on Diesel work.

Sensible and Latent Heats:

Sensible and latent heats must be carefully distinguished in studying the action of heat on matter. The term "sensible heat" is easily understood, but we may say, that sensible and latent heat represents latent and sensible work; that the former is actual, kinetic, heat energy, capable of transformation into mechanical energy, or vice versa, of masses, and into mechanical work; while the latter form is not heat, but is the equivalent of heat transformed to produce a visible effect in the performance of molecular, or internal as well as external work, and visible alteration of volume and other physical conditions.

It is seen that heat may become "latent" through any transformation which results in a defined physical change, produced by expansion of any substance in consequence of such transmutation into internal and external work; whether it be simple increase of volume or such increase with change of physical state.

Latent Heat of Expansion:

The latent heat of expansion may be defined as that heat which is demanded to produce an increase of volume, as distinguished from that untransformed heat which is absorbed by the substance to produce elevation of temperature. The latent heat of expansion may, by its absorption and transformation, and the resulting transforming of internal and external work, cause no other effect than change of volume, as e. g. when air is heated; or it may at the same time produce an alteration of the solid to the fluid, or of the liquid to the vaporous state. The specific heat of constant volume, no molecular or other work being done, measures the heat untransformed, and, as sensible heat producing rise in temperature. The specific heat of constant pressure measures the sum of latent and sensible heats, when a gas is heated, and no alteration of physical state can occur. It usually is assumed to include both internal and external work, as well as sensible heat; but where used in an unaccustomed sense the conditions of the case are always stated.

The Latent Heats of Fusion and Vaporization:

The latent heats of fusion and vaporization measure the quantities of heat transformed in these changes of physical state. In the first of these two cases the work done is mainly internal; in the second the internal work performed is much greater, but is not so enormously in excess of the amount of external work done; and the higher the pressure under which vaporization takes place, the larger proportionately the measure of external work and of the heat demanded for its performance.

Conduction:

Conduction is the method of transfer of heat flow from part to part in the same body, or from one to another of bodies in contact. These phenomena are not precisely the same. The flow of heat from a hot to

a cold body in contact depends not only on the conducting power of the two substances, but also, and often mainly, on the condition of the touching surfaces and the perfection of their contact. The rate of transfer within any given material depends solely on the variation of temperature along the line of flow, and on the character of the substance.

Calorific Values of Fuels:

The calorific value of fuel is the amount of heat, expressed in thermal units, evolved by the complete combustion of a unit weight of the fuel in the oxygen. These values are determined by the use of an apparatus known as the "Calorimeter".

Values of Liquid Fuels:

It is but natural that the value of fuel differs with the amount of properties it contains. It should be understood that the fuel question of the Diesel engine is easiér solvable than any other type of machinery, in particular the steam engine. It is a fact, which has been demonstrated under the most expert observation of engineers in this country as well as in Europe, under different climatical conditions, that the Diesel engine will consume the cheaper kind of oils with astonishing results.

In many cases compounded oils, such as lard oil, vegetable oils of almost any known kind, in fact any oil with the flash-point of very low degree, sometimes intermixed with coal tar, obtained of coal of poor quality give the greatest results. In other words, the maximum results are obtained with the minimum amount of fuel value, resulting in low expenditure. To get a clear conception of the qualities of the usual known kinds of solid liquids, it will be beneficial to study a few ordinary liquids. The analysis, as expressed in the terms of the chemical language, is as follows:

C	Carbon
H	Hydrogen
N	Nitrogen
O	Oxygen
S	Sulphur
A	Ash

Coal Tar:

The value of coal tar as a fuel is usually very much lower than its value for other purposes, but as a fuel-medium for Diesel engines it is very valuable. The yield of coal tar varies with the kind of coal and with the methods employed. From about 4½ per cent to 6½ per cent of the weight of coal. It is lower in hydrogen and higher in carbon than crude oil, and therefore of a lower calorific value. Tar made from standard gas coal would have an ultimate analysis about as follows:

Carbon	89.21%
Hydrogen	4.95%
Nitrogen	0.11%
Oxygen	4.23%
Sulphur	0.56%
Ash	Trace

It has specific gravity of about 1.25, a gallon weighing 10.3 pounds. Coal tar may be burned if heated and strained, the same as other liquid fuels.

Oil Tar:

Oil tar is produced in ordinary gas apparatus, has a specific gravity of 1.15, is less sticky than coal tar, and can be transported, handled and burned like other oils. Its analysis is about as follows:

Carbon	92.7 %
Hydrogen	6.13%
Nitrogen	0.11%
Oxygen	0.69%
Sulphur	0.37%
Ash	Trace

It is important that the fuel oil burned in the Diesel engine should be carefully examined and should be free from incombustible solids. The oils should be mobile at 0 degrees C., as it is heavy and rather viscous, or contains considerable proportions of asphaltum or paraffine, it will become sluggish and stiff at low temperatures and considerable heat will have to be supplied to warm it before it can be run to the engine.

If it is necessary to use very heavy or viscous oils the engine should first be warmed by running on a lighter fuel oil, and the heavy oil introduced after the engine is running well and warmed up. This process should be reversed in shutting down, in order to wash the heavy oils out of the fuel valves and small passages and pipes.

If the heavy oil is fed into the cylinders without first being preheated and while the engine is cool, it will tend to form a deposit on the cylinder head, and also will give off petroleum vapor, which requires a greater amount of oxygen for combustion than what is contained in the volume of air in the cylinders.

For proper combustion the oil should be free from water, grit and other solids. At least 80% of the oil should distill over at 350 degrees C. Also the oil should not contain more than 4 % of material insoluble in xylene as a large proportion of insoluble material will tend to form coke in the cylinders.

The presence of sulphur in small percentages (1%) especially if the oil be free from water, is not injurious, but in larger content will cause corrosion of the working parts. The best results have been obtained from California oils, if the lighter oils, such as gasoline and naptha,

which are of a greater commercial value, have been drawn off by distilling, and the asphaltum contents reduced to about 20 or 25%. Of this subject we will later dwell on, as the fuel question is of vital importance.

British Thermal Units:

A British Thermal Unit is the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit, at or about 39.1 degrees Fahrenheit, and represents a mechanical energy of 778 foot pounds.

Hydro-Carbons:

A chemical compound, or rather a chemical combination commonly found in large percentage in petroleum and coal-tar as well as practically all our vegetable oils, composed of hydro-carbons or compounds closely related thereto.

The Composition of Water:

Water is a neutral compound, exhibiting, when pure, neither acid nor alkaline reaction; but so freely does it dissolve substances with which it is brought in contact, that it is rarely found in nature absolutely free from either acidity or alkalinity. Its presence is essential to nearly all the chemical operations of nature, as well as in the artificial product of solid liquids.

The fluid may be decomposed in either of several ways, as by heat alone, a process of "dissociation" of its elements taken place at between 2000 degrees and 4000 degrees Fahr. (1100 degrees to 2200 degrees C.) or by voltaic current, and by the action of various metals or metalloids at high temperatures, when the substance employed has a strong affinity for the oxygen, as have carbon, iron, etc.

Water is found wherever hydrogen is burned, in air or oxygen, either alone or in combination with other elements. It enters into combination with many other substances and as water of crystalization, for example, often influences the character of the compound to a very important degree.

Composition of Sea Water:

Sea water is a mineral water, strongly saline, considerably chlorinated, and slightly alkaline. The composition of the water of the ocean differs very slightly in different localities. It contains about 1/32 of its own weight of salts, mainly common salt, with various other chlorides and bromides, and some gases. Deposits from sea water and from any other water containing solid matter either in solution or suspended, will always occur on evaporating the water; and these deposits form the incrustation and sediment which endanger the passageways, or commonly known as water-jacketing, on Diesel engines, and may lead to serious consequences when not properly attended to.

CHAPTER II.

THEORY

GAS

A gas is a substance whose molecules are repellent to each other, or in other words, has a tendency to separate.

When a quantity of gas has definite pressure and volume at a certain temperature, it is said to be in a certain **state.** Now if any outside agent should affect the gas in such a manner as to change the pressure, volume or temperature, it is said to undergo a **change of state.**

Suppose a cylinder is fitted with an air-tight piston, which is forced down quickly, compressing the air, the work done by the piston in compression is given off to the gas in the form of heat, consequently the temperature will rise in proportion to the pressure exerted on it, which change of state is known as **adiabatic compression.**

Then adiabatic expansion and compression is when the temperature of gas changes as the volume increases or decreases.

If in the same cylinder the piston were pushed down very slowly allowing the heat to pass through the cylinder wall as soon as forced, the temperature on the inside would be the same as that on the outside, and would be known as **isothermal expansion.**

Then isothermal expansion or compression is the increase or decrease of a volume of gas by compression or expansion without producing a change in temperature.

In the accompanying diagram (figure a) let the **axis of volume B** represent cubic feet of gas and **axis of pressure** A represent the pressure per cubic foot; suppose you had 10 cubic feet of gas at a pressure of 10 pounds absolute and compressed it to 8 cubic feet; the product of the pressure and volume would be 10 x 10=100; as it is compressed isothermally the product of the pressure and volume must remain unchanged, consequently $100 \div 8 = 12\frac{1}{2}$ pounds. If we compress the gas to 6 cubic feet we would have $10 \times 10 = 100$; $100 \div 6 = 16\frac{2}{3}$ pounds.

If compressed to 4 cubic feet we would have $10 \times 10 \div 4 = 25$ pounds, and if compressed to 2 cubic feet it would equal $10 \times 10 \div 2 = 50$ pounds, etc. If the 2 cubic feet of air was expanded back to the 10 cubic feet and heat were added to the air to keep it at the same temperature, it will pass through the same stages on the curved line as it did in compression.

In practical work on Diesel engines, ammonia compressors, etc., we have to deal with adiabatic expansion and our compression which not only contains the increase of pressure to the decrease of volume, but an additional pressure caused by the increase of temperature which makes it a little more difficult to figure. If no heat were lost through cylinder walls, leaks, etc., constant 1.405 could be used, giving an accurate pressure at any stage of the compression, but as engines differ considerably in heat lost only an approximate constant can be given, which is 1.26. In nearly all experiments on gas and oil engines it has given satisfactory results.

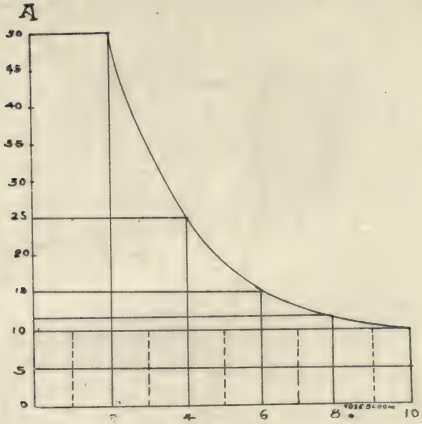


Figure (a).
Demonstration of Isothermal Expansion

We will suppose we have 10 cubic feet of air in diagram (b) under a 10 pound pressure, then we compress it to 8 cubic feet; now we have $12\frac{1}{2}$ pounds pressure plus an additional pressure caused by the heat of compression, consequently logarithm must be used to find the value of the constant, which is PV 1.26 (P=pressure V=volume) then log P=1+1.26=2.26 log of pressure, then log 10=1; 1×2.26=2.26, log of constant; constant is 182. Then the product of volume of gas and pressure must equal 182. Now if the volume is compressed to 8 cubic feet and if V 1.26 equals 182; P×8 1:26, then log 182=2.26007—1.26×.90309=1.13789 or log of pressure, then P=13.73 pounds. If the volume is compressed to 6 cubic feet we have constant 182—(6) 1.26 or log 182=2.26007—1.26 × .77815=1.27961 log of pressure; then P=18.6 pounds. If compressed to 4 cubic feet it would be log 182 =2.26007—1.26 (log 4) or 2.26007—1.26 × .60216=1.50148 log of pressure; then P=31.74 pounds. If compressed to 2 cubic feet the pressure

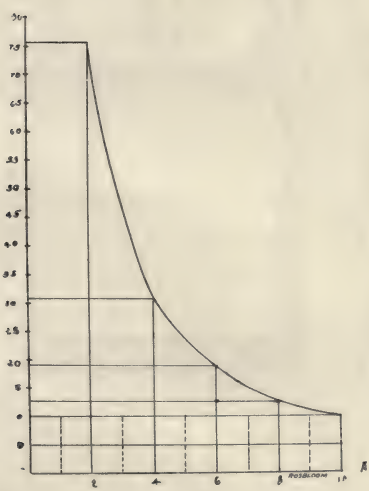


Figure (b).
Demonstration of Adiabatic Expansion.

would be log 182=2.26007—1.26 x .30103=1.88078 log of pressure; then P=75.96 pounds, etc.

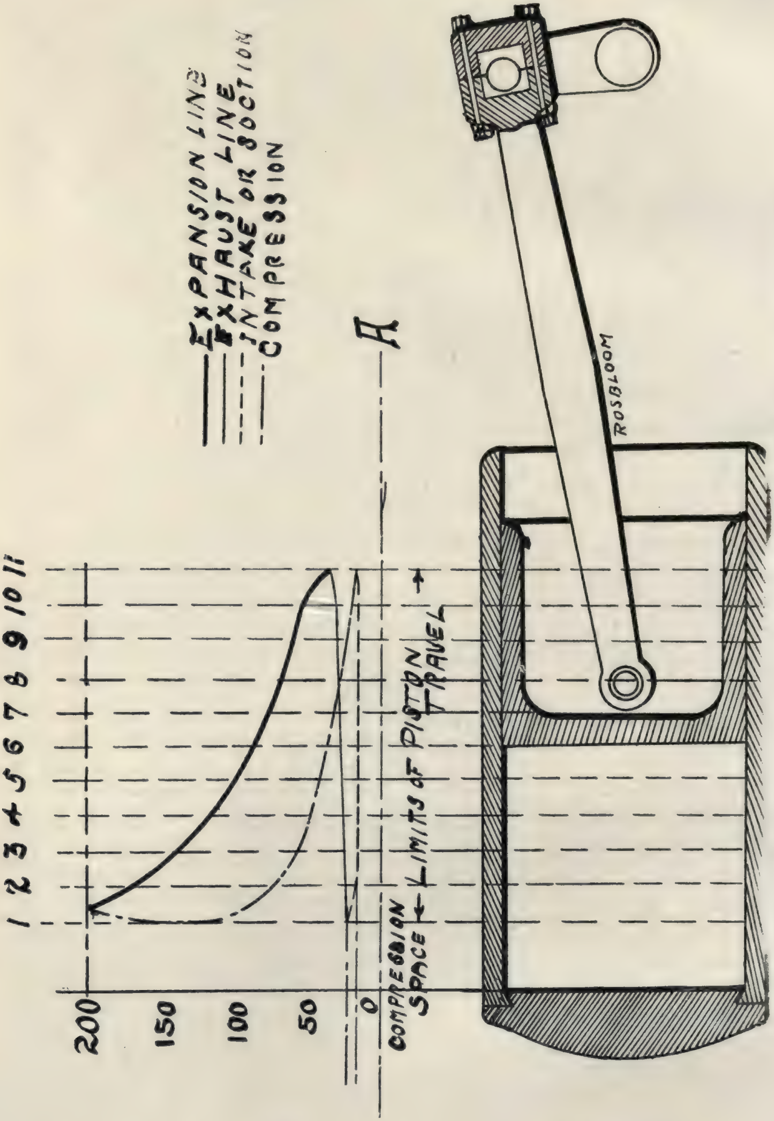


Figure (c). Practical Demonstration of Indicator Card Calculation.

The only way to find the action and pressure of gas in the cylinder is to attach an instrument to it, called the indicator, and take an indicator card, which records the pressure on the piston at all points in the cylinder.

Suppose in figure (c) that ignition takes place when the pressure reaches 100 pounds and rises to 200 pounds, as the piston passes dead center on its working stroke the gas continues to burn until the piston is half way to ordinate 2, at which point combustion has been completed and the pressure gradually decreases until it reaches the point A, then the exhaust valve opens and the piston on its return stroke (see exhaust line on diagram) forced the burnt gases out; as the piston moves downward on the second revolution a new charge of gas is drawn in (see suction line on diagram) forces the burned gases out; as the piston moves downward on the second revolution a new charge of gas is drawn in (see suction line on diagram); then on the return stroke the fresh charge is compressed (see compression line on diagram) to 100 pounds at which place it is ignited again.

A little study of the above diagram will give the reader a good idea of the relation that exists between the indicator card and the pressure acting on the piston in the cylinder.

Suppose AB represents the **axis of pressure** and BC represents the **axis volume of cylinder,** then any distance measured from AB in the direction and parallel with BC is the abscissa of that point and all the lines starting from axis BC and parallel with axis AB are known as the ordinates.

To make it more clear consider distance F as clearance volume, to the same scale as that of L, which represents cylinder volume. The axis BC may either represent atmospheric pressure or line of no pressure, that is, absolute vacuum, usually BC will represent atmospheric pressure.

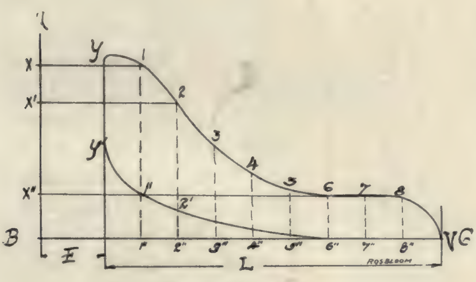


Figure (d).
Demonstrating Expansion of Gases in Cylinder.

In figure (d) the abscissa of the point is xl, and the ordinate of the point is 1″ 1, then the ordinate and abscissa of 1 are 1″ 1 and x 1 respectively, of 2 are 2″ 2 and x 2, etc.

The line yv represents the expansion of the gas, or in other words the pressure per square inch acting on the area of the piston throughout the full stroke (the diameter L). When ignition takes place the pressure acting on piston is equal to tp yw pounds; as the piston moves forward to line 1″ 1 the pressure on the area of the piston has decreased to x 1 pounds, etc., until piston reaches end of stroke at v and exhaust escapes from cylinder, consequently the amount of work performed is equal to the number of pounds pressure acting on the area of piston through the full length of stroke. In order to find the area of indicator card it is necessary to divide it

into a number of ordinates as shown in figure (d) which are 9, then add the length of all of them together and divide by number of lines, this will give the average heights of the diagram which is called **mean ordinates**, then multiply the mean ordinate by the length of stroke (distance L), this will give the mean effective pressure (abbreviated to M. E. P.) acting on the working stroke, but as a small amount of the power given off in the working stroke, will be consumed in compressing the new charge of gas in the return stroke, therefore it will be necessary to subtract the area of compression from the area of expansion, in order to find the effective foot pounds of work performed in one cycle. The area of compression is found by the same method as in expansion as follows: Ordinate y'w, 1' 1", 2' 2", etc., divide by the number of measurements taken and multiply by x" 6 (length of compression), which will give the power lost in compression.

In figure (d) we will suppose that ignition took place when the pressure reached 450 pounds; (the piston would almost be on dead center) the explosion causing the pressure to raise to a little more than 550 pounds before the piston started downward on the working stroke, as the piston moved downward the decrease of pressure is recorded on the indicator card until the exhaust valve is opened and the pressure inside the cylinder equalizes with the atmospheric pressure on the outside. Then we take the card and find the length of ordinates as described in figure (d), which are 13 10-16 inches, changing 13 10-16 inches to 16th., we would have 16 × 13 + 10 or 218 sixteenths, which is the length of the 12 ordinates; then 218 ÷ 12 = 18⅙ sixteenth, or the average height of 1 ordinate.

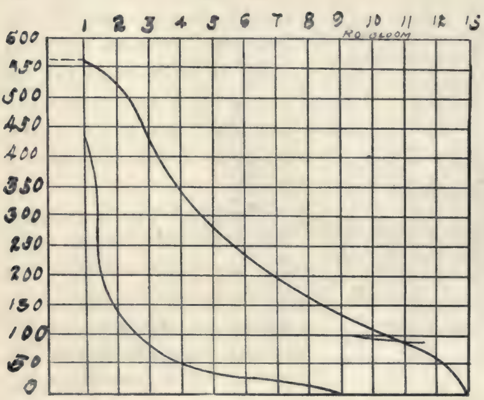


Figure (e).

Practical Demonstration of Indicator Card.

If 2 10-16 inches = 600 (the scale of pounds is found by the number of springs used); 1/16 = 600 ÷ 42 or 14 2/3 pounds, then 18⅙ × 14⅔ = 266 4/9 pounds to the square inch acting on area of piston throughout the full stroke, which is 12 inches —then 12 x 266 4/9 = 3197⅓ inch pounds of work performed; now we measure the mean ordinates of the compression, which are 4 inches; then 4 ÷ 8 = ½ inch—length of mean ordinate; then ½ × 16 or 8 sixteenths. 8 × 14⅔ = 117⅓ pounds. Therefore the compression averages 117⅓ pounds to the square inch for 8 inches; then 8 × 117⅓ = 933⅔ inch pounds of work performed in compression. Then 3197⅓—9332⅔ = 2263⅔ inch pounds of

effective work performed in one cycle. In order to find the M. E. P. we divide 2263⅔ by the length of stroke in inches or 2263⅔ ÷ 12 = 188.63 pounds.

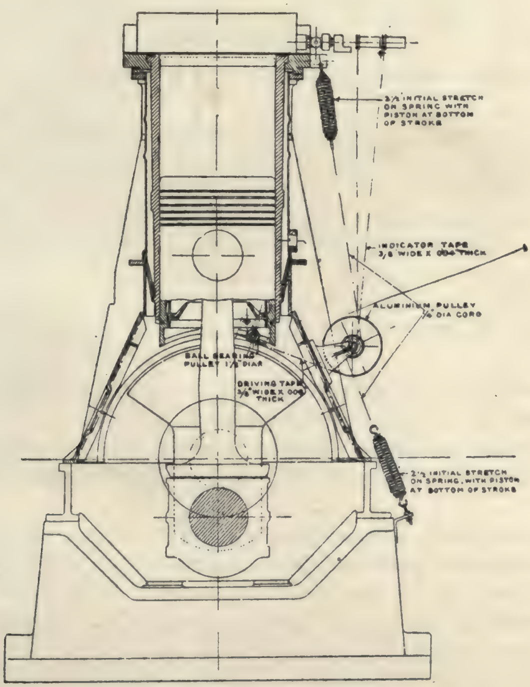
NOTE—Some figures may be eliminated by subtracting the area of compression from area of expansion and dividing by length of stroke, which will give the measurement for the scale of pounds.

PROPER MANAGEMENT AND NECESSARY PRECAUTIONS IN USING THE INDICATOR

(1) Before using the indicator give it a proper inspection. Ascertain the condition of the indicator. Any defective valve or leaky mechanical contrivance should be remedied before using the same.

(2) Add a small quantity of oil in drum spindle and piston. Use only mineral oil. Oil of heavy viscosity should never be used. It causes gumming and the result will be a retarding of the instrument with consequently bad results.

(3) The advisability of properly rigging the instrument to the engine cannot be overestimated. Previous to operating the engine obtain the correct length of cord.



Practical Application of Indicator.

(4) While engine is being operated with attached cylinder never alter any parts on indicator parts. It may result in improper termination of Cards.

(5) The paper on the drum must be properly placed. See that the paper clips are working perfectly.

(6) The pencil should be fairly sharp and brought to bear against the card very lightly.

(7) Any assembling of saturation inside the instrument should be avoided.

(8) When using Thompson Indicator be sure that the three-way cock has the proper amount of opening. Any throttling of power may have the result of imperfect recording.

MEASUREMENT OF INDICATED HORSE-POWER WITH THE INDICATOR

The indicated horse-power, or the rate at which an engine receives mechanical energy, is measured by the mechanical energy imparted to the piston per minute in foot-pounds, divided by 33,000.

The work done per stroke of L feet by the pressure on a piston whose area is A square inches is

$$p\ A\ L \text{ foot-pounds},$$

p being the mean effective pressure per square inch in pounds upon the piston.

If N is the number of power strokes per minute, i. e., the number of strokes of the piston when the driving pressure p is acting, then the indicated horsepower =

$$\text{I. H. P.} = \frac{P\ L\ A\ N}{33{,}000}$$

This may be applied to every cylinder when the engine is working uniformly.

The quantity N in the above formula may be obtained by some form of tachometer or by some form of a counter.

The area of the piston is best obtained by removing the cylinder cover and measuring the diameter of the cylinder with a micrometer gauge.

It is necessary to measure the diameter of the cylinder and piston rod at a number of positions, and take the mean of each of them, as these often wear out of truth.

The length of the stroke can be obtained by marking the cross-head guide at the extreme ends of the stroke.

The mean effective pressure is obtained from the indicator diagram. A comparison of previously taken indicator cards of the engine will give

accurate performance of mechanism and determine faulty operation. In other pages of this book a thorough explanation is given as to the measurement requirement properly ascertaining the function of the engine.

DEFINITION OF "CARNOT" AND "OTTO" CYCLE

The definition of cycle may be used to indicate a period of time in which a series of events repeat themselves; a recurring series of events or a series of operations terminating to its original state.

In the Carnot cycle its operation performed by perfect gas under perfect mechanical efficiency proves that thermodynamic—heat converted into mechanical work—efficiency of this engine is the highest that can be obtained by the use of any substance or combinations of substances in any engine working in any other cycle between the identical limits of existing temperature.

The practical engine as it improves approaches this efficiency, but can never attain it. In other words, the nearer the efficiency of any heat engine is to that of the Carnot cycle efficiency, the nearer it is to its highest attainable limit of perfection.

As will be seen in the subject dealing with the isothermal and adiabatic expansion, the heat must be lowered by an adiabatic expansion in which the heat that disappears does so in doing its mechanical work. We begin to realize that the compression temperature on a Diesel engine acting adiabatically, in order that the heat received as heat may be received at the highest possible temperature making it possible to accomplish the desired results.

In theory internal combustion engines work on either of the Otto or the Carnot cycle. In defining this theory in the laws of thermodynamics alluding to internal combustion engines, we must first understand that all engines used for generation of power are heat engines.

To establish an ideal standard of comparison in different types following exclusive processes of operation the term cycle is applied. When giving this subject a little thought we soon conclude that the cycle of operation, as the indicator application will convince us, is distinctly different from that of the Diesel.

The Diesel engine, which is a "constant pressure" engine, or to be plain, in which all the heat is taken to generate its power while the pressure remains "constant" in the cylinder and its rejection occurs in identical condition, follows an exclusive "Diesel" cycle peculiar to this engine.

In contrast to this constant pressure cycle we have the "constant volume" cycle, which we pleased to call the "Otto" cycle. In the study of Diesel engineering we find that this constant volume only exists in the gas or gasoline driven engine, where a constant volume establishes the "volumetric efficiency" of the engine. This volumetric efficiency is the fundamental principle of operation determining the results of power production of the engine itself.

HEAT

All bodies are supposed to be composed of minute particles—so small that they can scarcely be seen by a high powered microscope, they are called molecules. These molecules have weight and motion; in fact, the energy that any body possesses is due to the rapidity that these molecules vibrate to and fro. The state of any substance whether gaseous, liquid or solid, is determined by the attraction that these molecules have for each other, as follows:

In a gas, the molecules are said to be repellent to each other, that is, their adhesive qualities are entirely suspended, which causes them to travel away from one another in the direction of least resistance, thereby producing a state known as expansion—there is no limit to the expansion of an unconfined gas.

In a liquid body the adhesive qualities are only a slight degree greater than the cohesive qualities, thereby creating a state that permits the molecules to pass freely over or under one another in any direction, which acting with the laws of gravity, causes them to seek the lowest possible level. A liquid has no definite shape of its own, but assumes the shape of the object in which it rests.

In a solid the adhesive qualities are great enough to overcome the cohesive qualities to the extent of causing the mass to assume a definite size and shape, thereby forcing the molecules to remain in a fixed path of motion. A force of more or less degree is required to cause a solid to change its shape.

The vibratory motion of these bodies determines how hot or how cold the body is, for example:

When a liquid boils, it is the maximum stage of motion for the molecules and they separate from the main body of liquid and pass off into the air in the form of vapor or gas.

If you take a piece of iron and apply enough heat the molecules will move so rapidly that they travel outside of their fixed path of motion, losing their attraction for each other. Consequently the molecules will seek the lowest level, at which state it is said to be melted.

On the other hand, if we should extract enough heat from a gas the molecules would lose sufficient motion and condense into a liquid. If we continued to extract heat it would finally become a solid, such as ice, and as we abstracted heat the motion of the molecules would become less until they came to a state of rest, which would be 460 degrees below zero (Fahrenheit). So far this has been impossible, the lowest known temperature on record being in the neighborhood of 400 degrees below zero (Fahrenheit).

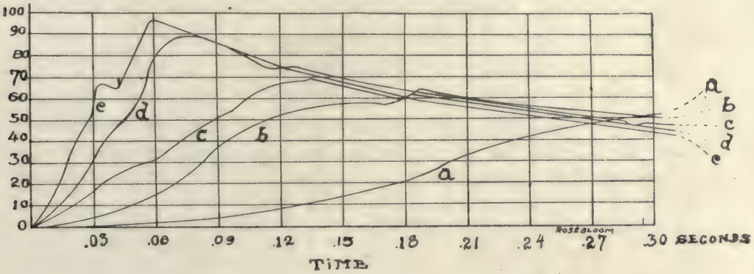
It can be readily seen that the temperature of any body is only a measurement of motion of the molecule. These measurements are taken by an instrument called a thermometer.

COMBUSTION

In the diagrams a, b, c, d, and e are pressures taken on a card by an instrument used for finding the effects of different mixtures of gas and air, when ignited in a cylinder at atmospheric pressure, that is, ignition taken place at the lower left hand corner and the pressure raises in proportion to the mixture of air and gas, which may be summed up as follows:

Diagram	(a)	(b)	(c)	(d)	(e)
Volume of air to 1 volume of gas	13	11	9	7	5
Time of explosiou: Second	.28	.18	.13	.07	.05
Guage pressure. Pounds per sq. in	52	63	69	89	96

In the table as shown here it can be seen that a mixture 5 of air to 1 of coal gas gives the best results. If we were to use 4 of air to 1 of gas the pressure would be less as there would not be sufficient oxygen to complete combustion, consequently the elements of gas control the amount of air to be used. Experiments have proven that one volume of



Experiments of Coal—Gas and Air.

air well saturated with gasoline mixed with 6 to 9 volumes of free air (depending upon the grade of gasoline) gives the greatest mean effective pressure.

All fuel oils and gases contain hydrogen and carbon, and are known as hydrocarbons, which when mixed with oxygen and burned the hydrogen and carbon seperate and unite with the oxygen, forming water (H_2O) and carbon dioxide (CO_2). If the carbon unites with only one part oxygen it forms another substance known as carbon monoxide.

The products of combustion are oxygen, carbon and hydrogen, combined with one part oxygen forms water.

The elements with their atomic weights usually found in fuel are as follows:

Elements:		Atomic Weight
Hydrogen	H	1
Oxygen	O	16
Nitrogen	N	14
Carbon	C	12
Sulphur	S	32

In the above table we find that the atomic weight of carbon is 12 and the atomic weight of oxygen is 16, then in carbon dioxide (CO_2) we have by weight 12 parts of carbon to 32 part of oxygen. In other words it requires 32 ÷ 12 or 2⅔ pounds of oxygen to one pound of carbon. As only 23% of the air, by weight, is oxygen, we have 2⅔ ÷ .23 or 16.6 pounds of air to supply the 2⅔ pounds of oxygen, which may be summed up as follows,

$$1\ C + 11.6\ \text{air} = 12.6\ \text{pounds} \text{ ---------------Mixture}$$

$$1\ C + \left\{\begin{matrix} 2.67\ O \\ 8.93\ N \end{matrix}\right\} = 12.6\ \text{pounds} \text{ -----------Elements}$$

$$\text{Carb. diox.}\left\{\begin{matrix} 1\ C \\ 2.67\ 0 \end{matrix}\right\} + 8.93\ N = 12.6\ \text{pounds----Products of Combustion}$$

In the above table 1 pound of C requires 11.6 pounds of air for combustion. The 2.67 pounds of O in the 11.6 pounds of air combines with the 1 pound of carbon forming 3.67 pounds of CO_2 and the 8.93 pounds of N pass off with the CO_2 without taking any part in the combustion.

In hydrogen the product of combustion is H_2O, by weight it is composed of 2 parts of H to 16 parts of O, then 16 ÷ 2 = 8 pounds of oxygen to unite with 2 pounds of hydrogen. As oxygen by weight equals 23% of the air, we have 8 ÷ .23 or 34.8 pounds of air to burn 2 pounds of hydrogen, which is summed up as follows:

$$2\ H + 34.8\ \text{air} = 36.8\ \text{pounds} \text{ ---------------- Mixture}$$

$$2\ H + \left\{\begin{matrix} 8\ \ O \\ 26.8\ N \end{matrix}\right\} = 36.8\ \text{pounds} \text{ -------------Elements}$$

$$\left\{\begin{matrix} 2\ H \\ 8\ O \end{matrix}\right\} + 26.8 = 36.8\ \text{pounds--Products of Combustion}$$

Suppose there wasn't sufficient oxygen to unite with the carbon to form complete combustion, then we have as stated before CO instead of CO_2, that is, by weight 12 parts of carbon to 16 parts of oxygen or 16 ÷ 12 = 1⅓ pounds of oxygen to 1 pound of carbon, which is summed up as follows:

$$1\ C + 5.8\ \text{air} = 6.8\ \text{pounds} \text{ ------------------Mixture}$$

$$1\ C + \left\{\begin{matrix} 1.33\ O \\ 4.47\ N \end{matrix}\right\} = 6.8\ \text{pounds} \text{ -------------Elements}$$

$$\text{Carbon monox.}\left\{\begin{matrix} 1\ C \\ 1.33\ O \end{matrix}\right\} + 4.47 = 6.8\ \text{pounds--Products of Combustion}$$

It is not customary to use the weight of gas in calculating the pounds of air required for combustion in gas engines, as gas is invariably meas-

ured in cubic feet, making it necessary to figure air by cubic feet, which may be explained as follows:

The combustible products of hydrocarbons are CO_2 and H_2O. It is evident that in each molecule of CO_2 it requires two atoms of oxygen to complete the combustion of 1 atom of carbon and for every molecule of H_2O it would repuire ½ atom of oxygen to complete combustion for 1 atom of hydrogen, which may be expressed in the formular of

$2C + \frac{H}{2}$ = number of atoms of oxygen required to burn any hydrocarbon.

In chemical theory a gas requires ½ as many volumes of oxygen as there are atoms of oxygen in the compound. Then the volume of oxygen required for complete combustion for any hydrocarbon may be found in the following formula:

$$\frac{2C + \frac{H}{2}}{2} = (C + \frac{H}{4})\ 4.76$$

As only 21% of air by volume is oxygen, we would have 1 — .21 or 4.76. The volume necessary for complete combustion of 1 volume of hydrocarbon, consequently we have: Volume $= (C + \frac{H}{4})\ 4.76$.

Example:

How many cubic feet of air would be required to burn one cubic foot of Hexane (C_6H_{14})?

Solution:

$$V = (C + \frac{H}{4})\ 4.76 \text{ or } V = (6 + \frac{14}{4})\ 4.76$$

then $V = 9.5 \times 4.76$ or 45.2 cubic feet of air.

To find the volume of air required for a gas containing a mixture of various hydrocarbons, use the above formula for each constituent of the gas and multiply the results by the percent and add together.

Example:

How many cubic feet of air would be required to burn gas composed of the following?

Constituents of Gas:

Propane C_3H_8 ------------------------------15%

Methane CH_4 ------------------------------75%

Butane C_4H_{10} ------------------------------10%

Solution:

$$V = (3 + \frac{8}{4})\ 4.76 \text{ or } V = 5 \times 4.76 = 23.8 \text{ cu. ft. of air}$$

$$V = (1 + \frac{4}{4})\ 4.76 \text{ or } V = 2 \text{ x } 4.76 = 9.52 \text{ cu. ft. of air}$$

$$V = (4 + \frac{10}{4})\ 4.76 \text{ or } V = 6.5 \text{ x } 4.76 = 30.94 \text{ cu. ft. of air}$$

Then

Propane = 23.8 × .15 = 3.57 cu. ft. of air
Methane = 9.52 × .75 = 7.14 cu. ft. of air
Butane = 30.94 × .10 = 3.09 cu. ft. of air

Total ------------------------13.8 cu. ft. of air. Answer.

RELATIVE WEIGHTS OF ELEMENTS

The usual weight definition of the atoms or the atomic weights of the elements are expressed in terms of the weight of an atom of hydrogen. Thus H = 1, O = 16, N = 14, C = 12, and S = 32.

Since equal volumes of gases contain the same number of molecules it follows that the approximate weights of equal volumes of gases will be the same as the relative weights of their molecules.

TABLE OF RELATIVE ATOMIC AND MOLECULAR WEIGHT

Hydrogen ------------------	Atomic Weight	Molecular Weight	
Oxygen ------------------	H = 1	$H_2 = 1 \times 2$	= 2
Nitrogen ------------------	O = 16	$O_2 = 16 \times 2$	= 32
Carbon ------------------	N = 14	$N_2 = 14 \times 2$	= 28
Sulphur ------------------	C = 12	------------	-----
Element or Compound	S = 32	------------	-----

COMPOSITION OF AIR

For an approximate combustion calculation the composition of atmospheric air may be taken as follows:

Oxygen------------------------23 per cent------------------21 per cent
Nitrogen ----------------------77 per cent------------------79 per cent

APPROXIMATE CALORIFIC VALUES OF THE COMMON COMBUSTIBLES

Combustible	Product of Combustible	Heat Evolved per Lb. of Combustible B. T. U.
Carbon (C)	Carbon Monoxide (CO)	4,450
Carbon (C)	Carbon Dioxide (CO_2)	14,540
Hydrogen (H)	Water (H_2O)	62,030
Sulphur (S)	Sulphur (S)	4,050
Carbon Monoxide (CO)	Carbon Dioxide (CO_2)	4,300
Ethylene (C_2H_4)	Carbon Dioxide (CO_2) and Water (H_2O)	21,500
Methane (CH_4)	Carbon Dioxide (CO_2) and Water (H_2O)	23,550

SOME FACTS ON COMBUSTIBLE SUBSTANCES

The principal components of liquid fuels are carbon, hydrogen, oxygen and nitrogen.

Oxygen does not burn, but it is a supporter of combustion.

The pressure of liquid at any point is equal in all directions.

There is an equal number of molecules in equal volumes of all gases at the same temperature and pressure.

Nitrogen will neither burn nor support combustion.

Water vapor, if present in large quantities, retards ignition and the propagation of explosion. Before an explosion can occur, or combustion, the vapor must be raised to the ignition temperature of the gas and on account of the high specific heat of water, considerable heat is thus absorbed.

The boiling point rises with increase of pressure and falls with decrease of pressure.

A cubic foot of dry air at 32° F. at sea level weighs 0.080728 lb.

Absolute zero is —459.4; above this temperature everything scientifically contains heat.

The density of a body depends both upon its mass and its volume.

Water is reduced only 0.00005 of its volume by a pressure of one atmosphere. A gas is reduced to one-half its volume by the same pressure.

Gases have no elastic limit. No amount of compression can permanently change their polume; they always return to their original volume when the distorting pressure is removed.

Velocity is the rate of motion.

Specific gravity is the given amount of water at 60 degree normal temperature. Other substances might be selected, but the most suitable standard is water, therefore it is used for the purpose of determining the density of solids and liquids.

The British Thermal Unit (B.T.U.) is a unit to measure the quantity of heat generated by the burning of substances. It is equivalent to the amount of heat required to raise the temperature of 1 lb. of water 1 degree of the Fahrenheit scale, or 1 B.T.U. is equivalent to 778 foot-pounds.

When heat is added to a body, whether solid, liquid or gaseous, the vibration of the molecules composing the body increases.

The centigrade scale differs from the Fahrenheit in making the freezing point 0° and the boiling point 100°, the space between being divided into 100 equal parts. This thermometer is the one in general use among scientific men.

Pure dry air is chiefly a mixture of oxygen, nitrogen and carbon dioxide, containing nearly four volumes or parts of nitrogen to one part of oxygen. Figures that are still more exact, and which are frequently used by the chemist when calculating the amount of oxygen in a given volume of air, are as follows:

	Per cent.
Carbon dioxide (CO_2)	0.23
Oxygen (O_2)	20.93
Nitrogen (N_2)	79.04

Oxygen is slightly soluble in water, 25 volumes of water will absorb one volume of oxygen.

The average pressure of the atmosphere at sea level is 14.7 lbs. per square inch. This is called the pressure of 1 atmosphere.

The weight per cubic foot of any gas at different temperatures and pressures can be found by the following formula:

Let W = weight in pounds;
V = volume in cubic feet;
B = barometric pressure;
S = specific gravity;
T = absolute temperature.

Hydrogen has no taste or color. The pure gas has no odor, though hydrogen as ordinarily prepared has a disagreeable odor, due mainly to impurities in the metals used. Hydrogen is the lightest known substance. Volume for volume, air is about 14.4 times and oxygen 16 times, and water 11,000 times heavier than hydrogen.

By specific heat is meant the quantity of heat necessary to raise the temperature of a substance one degree compared with the amount of heat necessary to raise the temperature of an qual weight of water one degree.

To measure the specific heat of a body the following will suffice as explanation: The quantity of heat absorbed by the cool body in heating = mass × change in temperature × specific heat.

The quantity of heat given out by the hot body in cooling = mass × change in temperature × specific heat.

Thus:

M = mass;
t = temperature change;
s = specific heat;
$Mts = M't's$.

It will be noticed that the heat absorbed by the cool body in heating is exactly the amount given out by the hot body in cooling.

In internal combustion engines the pressure in the cylinder is due to the action of the heat evolved during combustion. The capacity for heat of the combustible mixtures in the cylinder is, however, small, whilst the temperature of combustion is high. It follows, therefore, that any loss of heat will necessarily seriously affect the temperature of the products of combustion, with a consequent loss in efficiency.

The value of fuel depends upon the use that can be made of the store of latent energy which it contains.

BOYLE'S LAW

Boyle's Law states that for a given mass of gas, at constant temperature, the pressure varies inversely as the volume; or, using the letters P and V to represent pressure and volume respectively.

$$P = \frac{1}{V}, \text{ that is, the product } P \times V = \text{constant.}$$

FOR AIR: If V is the volume of 1 lb. of air in cu. feet and P is the pressure in lbs. per sq. foot, at the constant temperature of freezing water, 32° F.,

Then we have: U. V. = 26220.

CHARLES' LAW

Charles' Law states that when a given mass of gas expands under constant pressure, equal increments of temperature produce equal increments of volume, and it also states that all gases expand alike.

Thus if V_0 is the volume at zero temperature of a given quantity of gas expanding at constant pressure, and if V' is its volume at any other temperature T^1.

Then $V = V_0 (1 + aT^1)$, where a is a constant.

Thus when T^1 is in Centigrade units a is very nearly 1/273, and when T^1 is in Fahrenheit units a is very nearly 1/461.

JOULE'S LAW

Joule's Law states that heat and mechanical work are mutually convertible, a unit of heat being equivalent to a certain amount of mechanical work, called the **Mechanical equivalent of heat.**

This may be expressed in following: $w = J. H.$

where w = the amount mechanical work in work units,

and H = the quantity of heat in heat units,

J = the Mechanical Equivalent of Heat = 778 in Fahrenheit units.

The fact established by Joule is:

That a gas expands without actually doing external work and without taking in or giving out heat (i.e., without changing its internal energy), its temperature remains constant.

We may express this result in following:

Heat supplied = work done + increase in internal energy,

or: $H = W + (E - E_0)$.

CHAPTER III.

MISCELLANEOUS FORMULAS

To Find Indicated Thrust of a Propeller:

$$\text{Lbs. Thrust equals } \frac{\text{H. P.} \times 33000 \times \text{per cent utilized}}{\text{Pitch} \times \text{revol.} \times 100} = \text{lbs. thrust of propeller}$$

The resistance of the water varies as the square of the speed. The power required to overcome this resistance equals the cube of the speed.

To Find the Power of the Screw:

P equals Power required

P equals Pitch of Screw

L equals length of handle

W equals weight lifted

$$P = \frac{\text{weight} \times \text{pitch}}{\text{length} \times 2 \times 3.1416} = \text{power required.}$$

Displacement of Ship:

$$\text{A equals } \frac{\text{Tons} \times 35 \times 12}{\text{Inches sunk}} = \text{Sectional area of ship at water level}$$

$$\text{D equals } \frac{\text{Tons} \times 35 \times 12}{\text{Area}} = \text{Displacement in inches.}$$

$$\text{T equals } \frac{\text{A} \times \text{inches to sink}}{35 \times 12} = \text{Tons required to sink ship by amount of inches.}$$

$$\text{E equals } \frac{\text{Ton cargo} \times 35}{1 \times \text{b} \times \text{d}} = \text{Co-efficient of displacement.}$$

$$\frac{\text{Number of feet}}{6080} = \text{decimal part of 1 nautical mile.}$$

Suppose, for example, we have 12 feet to consider and wish to convert the 12 feet in decimal terms of 1 mile. Thus:

$$\frac{6080 \text{ ft.}}{12 \text{ ft.}} = .00197 \text{ nautical mile.}$$

The way this .00197 would be used in the kind of a problem this is referred to is as follows:

Suppose we have a propeller that has a pitch of 15 feet, and the loss in slip is 20 per cent, which is 3 feet per revolution, then the actual effective pitch is 12 feet, or, the propeller and vessel to which it is attached, will advance 12 feet per revolution.

Let us say that the propeller turns 60 times a minute and it is desired to find how far the ship has advanced in 4 hours, we will say:

If the propeller turns 60 times per minute, then, in 4 hours it will turn,

$$60 \times 60 \times 4 = 14{,}400 \text{ revolutions.}$$

The constant we will use, as before explained, is .00197, and so,

$$14{,}400 \times .00197 = 28.36+ \text{ miles (say 28.4 miles).}$$

To prove the constant .00197 is sufficiently close for all practical purposes, let us work out the same problem in the usual way, as follows:

Effective pitch of propeller=12 feet.

R. P. M.=60.

Hours run=4.

Then:

$$12 \times 60 = 720 \text{ feet per minute.}$$

$$720 \times 60 = 43{,}200 \text{ feet per hour.}$$

$$43{,}200 \times 4 = 172{,}800 \text{ feet in 4 hours.}$$

As there are 6,080 feet in one nautical mile, then as many times as 6.080 is contained in 172,800 is the number of miles (nautical) that the ship has advanced in the 4 hours:

$$172{,}800 \div 6{,}080 = 28.42 \text{ miles.}$$

The reason that, by the use of the constant, we get a slightly different answer, viz: 28.36+, is that the decimal value .00197 can be worked out further, hence the small difference as noted.

THE CIRCLE:

The circumference of a circle is equal to the diameter multiplied by 3.1416.

The area of a circle is equal to the square of the diameter multiplied by .7854.

To find the length of an Arc of a Circle: Multiply the diameter of the circle by the number of degrees in the arc and this product by .0087266.

To find the area of a Sector of a circle: Multiply the number of degrees in the arc of the sector by the square of the radius and by .008727; or, multiply the arc of the sector by half its radius.

THE TRIANGLE:

Varieties.—Right angled, having one right angle; obtuse angled, having one obtuse angle; isosceles, having two equal angles and two equal sides; equilateral, having three equal sides and equal angles.

The sum of the three angles of any triangle equals 180 degrees.

The two acute angles of a right angled triangle are complements of each other.

Hypothenuse of a right angled triangle, the side opposite the right angle, equals $\sqrt{\text{sum of the squares of the other two sides}}$.

To find the area of a Triangle: Multiply the base by half the height.

The Area of a Triangle being given to find the Length of the Base: Base equals twice the area divided by perpendicular height.

Area of a Triangle being given to find the Height: Height equals twice area divided by base.

QUADRILATERAL FIGURE:

To find the Area: Divide the figure into two triangles; the sum of the areas of the triangles is the area.

THE ELLIPSE:

To find the Area: Multiply the two diameters together and the product by .7854.

THE SPHERE:

To Compute the Surface: Multiply the diameter by the circumference and the product will give the surface.

To Compute the Total Volume: Multiply the cube of the diameter by .5236.

THE CYLINDER:

To Compute the Surface: Multiply the length by the circumference and add the product to the area of the two ends.

EQUIVALENT OF MEASURE (VELOCITIES AND ACCELERATIONS)

1 kine = 1 centimeter per second = 0.0328083 foot per second.

1 radian per second = 57.2958 degrees per second = 0.159155 revolutions per second.

1 gravity = 980.5966 centimeters per sec. = 32.1717 feet per sec.

1 foot pound = 13557300 ergs = 18325.5 gram-centimeters.

EQUIVALENT OF MEASURE (MILES, ETC.)

1 Yard, U. S.	1.0000029 yard British
1 Yard, British	0.9999971 yard U. S.
1 Chain, Gunter's	100 links
1 Link	7.92 inches

1 Cable Length, U. S. = 120 fathoms = 960 spans = 720 feet = 219.457 meters.

1 League, U. S. = 3 statute miles = 24 furlongs.

1 International Geographical Mile = $1/15°$ at Equator = 7422 m = 4.611808 U .S. statute miles.

1 International Nautical Mile = $1/60°$ at meridian = 1852 m = 0.999326 U. S. nautical miles.

1 U. S. Nautical Mile = $1/60°$ of circumference of sphere whose surface equals that of the earth = 6080.27 feet = 1.15155 statute miles = 1853.27 meters.

1 British Nautical Mile = 6080.00 feet = 1.15152 statute miles = 1853.19 meters.

TO FIND THE HORSE POWER REQUIRED TO DRIVE A SHIP THROUGH THE WATER AT A GIVEN SPEED

$$\text{H. P. equals } \frac{\text{Area of immersed midship section} \times \text{knots per hr.}}{600} =$$

Horse Power required.

Note: The resistance varies as the square of the speed.

Valve Formula:

Half the travel of the valve-lap equals the greatest opening.

Greatest opening $\times$ length of port equals Area of opening.

Lap plus lead plus exhaust lap equals exhaust opening

$$\frac{\text{Twice the lap plus lead}^2 \times \text{stroke equals Point of cut-off from end of stroke.}}{\text{Travel}}$$

Convenient Formulas:

$$S \text{ equals } \frac{C \times P}{p} = \text{length of stroke.}$$

$$C \text{ equals } \frac{S \times P}{P} = \text{point of cut-off.}$$

$$P \text{ equals } \frac{S \times P}{C} = \text{absolute pressure.}$$

$$p \text{ equals } \frac{P \times C}{S} = \text{terminal pressure.}$$

R equals S $\div$ C = ratio of expansion.

Pressure on Guide:

$$P \text{ equals } \frac{\text{Area of piston} \times P \times \text{length of crank}}{\text{Area of guide} \times \text{length of connecting rod}} = \text{pressure per sq. in. on guide.}$$

Crank Pin:

$$P \text{ equals } \frac{\text{Area of piston} \times \text{pressure}}{\text{Diam.} \times \text{length crank pin}} = \text{pressure per sq. inches on pin.}$$

Pressure:

P equals Pressure above atmospheric pressure when denoting bursting or safe working pressure, gauge, or safety valve pressures of tank.

P equals Absolute pressure when denoting engine pressures, strains on shafting, etc.

1. **Example:** A propeller is 9′ in diameter, 20″ across the blades, and the forward corner is 9½″ in advance of the after one. What is the pitch?

p = piece of pitch;

c = piece of circumference;

P = whole pitch;

C = whole circumference;

20 inches piece of thread;

9 5/10 piece of pitch.

400 minus 90.25 equals 309.75

9 × 12 equals 108″ diameter of wheel, (12″ = 1 foot)

3.1416 × 108″ equals 339.2928 inches whole circumference,

339.2928 inches × 9.5 equals 3223.2816 inches.

3223.2816 inches divided by 12 equals 268.6068 ft.

268.6068 divided by 17.6 equals 15.26 plus ft.—pitch.

$$P \text{ equals } \frac{C \times p}{c} \text{ or}$$

as c = to C as p = to P

20″ × 20 = 400″	400.00
9.5″ × 9.5 = 90.25	90.25
309.75 = 17.6″ part of circumference.	309.75

Answer: 15.26 plus ft. pitch.

2. **Example:** The pitch of a propeller is 18 ft. and makes 70 r. p. m. what is the speed of the ship in knots per hour allowing 20 per cent off for slip?

C = constant 6068 ft.

P = pitch of wheel;

R = revolutions per minute

S = percentage of slip;

K = knots; (60 minutes in 1 hour)

6068 ft. equals 1 knot.

$$K \text{ equals } \frac{P \times R \times 60 \times S}{C}$$

Efficiency:

A crude definition of efficiency is:

$$\frac{\text{What you get}}{\text{What you paid for it}}$$

The Mechanical Efficiency of the Engine is:

$$\frac{\text{B. H. P.}}{\text{I. H. P.}}$$

Apparent Slip:

The definition of this term means following:

A ship which is driven at an abnormal speed of V knots per hour by a propeller having a pitch of p feet, and making r revolutions per minute, the apparent slip is the quantity computed by the equation

$$s_1 = \frac{pr - 101.3\,V}{pr}$$

Real and Apparent Slip:

The slip of propeller may be defined in following:

$$s = \frac{pr - 101.3\,V\,a}{pr}$$

Where V a is the speed of the ship in knots per hour, p is the pitch in feet and r is the number of revolutions per minute.

Pitch Ratio and Slip:

The importance of determining the pitch ratio, where a change of propellers necessitates the accuracy of the type desired may be explained in following:

For large ships the pitch ratio usually range from 1.0 to 1.5 and the apparent slip from 0.10 to 0.20; both pitch-ratio and slip increasing with the speed-length-ratio. The efficiency of the full-faced type ranges from 0.45 to 0.75, increasing with the pitch ratio, and being larger for narrow blades and for propellers with few blades (three or two). The variation in this case for a given type of propeller is not large and can be known approximately in advance. For a given range of slip the efficiency changes but little, but there is an appreciable falling off for large slips. These conditions vary somewhat for the various pitch-ratios.

Number of Propellers:

The efficiency establishment in difference of single or twin propulsion, depends a great deal on prevailing conditions. The differences are not large and any of aforementioned type may be of equal efficiency. A single engine is, of course, simpler and cheaper than two engines, taking in consideration the initial expenses. For moderate powers and speeds a single screw will be chosen unless there are distinct advantages otherwise, such as the elimination of a heavy weight main engine to be substituted with twin engines, which are as easily operated as the single engine, owing to the facility afforded in managing Diesel power.

Thrust Computation:

To compute the thrust per square inch we may first find the effective horsepower by multiplying the indicated horse-power by the coefficient of propulsion—from 0.5 to 0.65. The effective horse-power may be multiplied by 33,000 to find the foot-pounds per minute, and this quantity divided by the speed of the ship in feet per minute (101.3V) will give the tow-rope resistance; this last quantity must be divided by 1 - + to find the thrust of the propeller; so that

$$\text{Thrust} = \frac{33000 \text{ E. H. P.}}{101.3 \text{ V } (1 - \text{t})}$$

in which V is the speed of the ship in knots and t is the thrust deduction (about 0.1).

Shaft Diameters:

FOUR CYCLE DIESEL ENGINE AT 120 R.P.M.:

6 Cylinders, 26″ diameter × 42″ stroke, 500 lbs., initial pressure.

$$\text{Diameter Shaft} = \text{constant} \sqrt[3]{\frac{\text{cyl. diam.}^2 \times \text{press.} \times \text{stroke} =}{\text{f (shaft stress)}}}$$

$$.97 \sqrt[3]{\frac{26^2 \times 500 \times 42}{7500}} = \underline{\underline{11.85''}}$$

FOUR CYCLE DIESEL ENGINE AT 90 R.P.M.:

6 Cylinders, 30″ × 48″, 500 lbs. initial pressure.

$$\text{Diameter Shaft} = .97\sqrt[3]{\frac{30^2 \times 500\ 48}{7500}} = 13.75''$$

TWO CYCLE DIESEL ENGINE AT 115 R.P.M.:

6 Cylinder, 22″ x 32″, 500 lbs. pressure.

$$\text{Diameter Shaft} = 1.04\sqrt[3]{\frac{22^2 \times 500 \times 32}{7500}} = 10.48''$$

TWO CYCLE DIESEL ENGINE AT 90 R.P.M.:

6 Cylinders, 24″ × 38″, 500 lbs. initial pressure.

$$\text{Diameter Shaft} = 1.04\sqrt[3]{\frac{24^2 \times 500 \times 38}{7500}} = 11.75''$$

2 — 4 Cycle Diesel Engines running in parallel at 150 R. P. M. with single Reduction Gear to one Main Shaft running at 85 R. P. M. Ratio of gearing 1.75 to 1, single impulses.

8 Cylinders, 17″ × 27″, 500 lbs. initial pressure.

The two engines form pratically one 16 Cylinder Engine. The twisting movement given under the root sign under 7 is to be multiplied by the gear ratio so that

$$\text{Diameter Shaft} = \text{constant}\ \sqrt[3]{\frac{D^2 \times P \times S \times 1.75}{7500}}\quad \text{Constant for 16 Cylinders.}$$

Four Cycle is 1.1.

$$\text{Diameter Shaft} = \sqrt[3]{\frac{17^2 \times 500 \times 27 \times 175}{7500}} = 10.6''$$

Engines with smaller cylinders running at higher revolutions and having consequently a greater gear ratio will have line shafts in diameter than the above engine.

(Note: Aforementioned formulas are from "Compilations by Joseph Hecking, Technical Staff, American Bureau of Shipping).

Strength of Seams:

P equals pitch;
D equals diameter of rivets;
T equals thickness of plates in inches;
A equals area of rivets;
N equals No. of rows.

(a) $$\left.\frac{P - D}{P}\right\} \times 100 \text{ equals } \left\{\begin{array}{l}\text{Percentage of strength of rivets as}\\ \text{compared with the solid plate.}\end{array}\right.$$

(b) $$\left.\frac{A \times N}{P \times T}\right\} \times 100 \text{ equals } \left\{\begin{array}{l}\text{Percentage of strength of rivets as}\\ \text{compared with the solid plate.}\end{array}\right.$$

Engine Formula: (Slow-speed heavy duty types)

The stroke equals the mean diameter of cylinders.

D divided by 10 equals diameter of piston rod;

D divided by 14 equals diameter of piston rod at bottom of thread.

D divided by 20 equals diameter of connecting rod bolts (2).

$$\frac{D \times \text{stroke}}{9} = \text{diameter of crankshaft journals.}$$

$$\frac{D \times S}{10} = \text{diameter of tunnel shaft journal.}$$

P = pressure in cylinder
L = length of stroke
A = area of cylinder
N = number of revolutions per minute
C = number of cylinders

For single acting two-cycle type, which has one impulse for each revolution, use the following formula:

$$\text{I. H. P.} = \frac{P\ L\ A\ N}{33000}$$

For four-cycle, which has an impulse on alternate strokes, use the following formula:

$$\text{I. H. P.} = \frac{P\ L\ A\ N}{2 \times 33000}$$

Note: Above formulas give horse-power of one cylinder. To ascertain the horse-power on multi-cylinder engines, multiply by number of cylinders.

Formulas for Brake Horse-power:

$$\text{For two-stroke cycle engine B. H. P.} = \frac{C\ L\ A\ N}{750}$$

$$\text{For four-stroke cycle engine B. H. P.} = \frac{C\ L\ A\ N}{1000}$$

Or by following: (Four-cycle engine single acting)

$$N = \frac{P \times s \times n}{880}$$

In the case of the two-cycle engine, single acting:

$$N = \frac{P \times s \times n}{500}$$

Where

N = B. H. P.

n = revolutions per minute

P = piston area in sq. in.

s = piston stroke in feet.

Estimated Indicated Horse Power Formula:

Values for the accuracy of the engine performance can be given only when indicator cards are taken. Indicated Horse Power is equal to $P \times L \times A \times N \div 33{,}000$,

where—

P = mean effective pressure in pounds per square inch on the piston;

L = length of piston stroke in feet;

A = area of piston in square inches;

N = number of working strokes per minute;

33,000 = number of foot pounds of energy expended per minute to make one horse power.

Mean effective pressure should be measured on indicator cards by an average planimeter. Mean effective pressure between exhaust and suction strokes in a four-cycle engine means a thermo-dynamic loss of energy from the piston. It should be deducted from the mean effective pressure between compression and working strokes before computing net power.

Ratio of Air Supplied to Air Used Formula:

Following formula may be applied:

$$R \text{ (ratio)} = \frac{N}{N - (3.8 \times 0)}$$

wherein

N = per cent of nitrogen in the exhaust gases:

O = per cent of oxygen in the exhaust gases

3.8 = the approximate ratio of nitrogen to oxygen in fresh air.

All these quantities refer to volumes of gases with all saturated properties eliminated. The per cent of nitrogen is assumed equal to the difference between 100% and the total carbon dioxide and exygen. oxygen.

Thermal efficiency of Engine, Fuel to Piston Formula:

E (efficiency) = B. t. u. equivalent of net indicated power per hour ÷ B. t. u. supplied in fuel per hour. (The B. t. u. equivalent of one-horse-power-hour is taken as 2545, which corresponds to 778 foot-pounds as the equivalent of one British thermal unit.)

B. t. u. in Brake Horse Power per Hour Formula:

The B. t. u. in brake horse power per hour = brake horse power × 2545.

B. t. u. in Jacket Cooling Water per Hour Formula:

The B. t. u. in jacket cooling water per hour = pounds of water flowing per hour × temperature rise in degrees Fahrenheit.

B. t. u. in Cylinder Water per Hour Formula:

The B. t. u. in cylinder water per hour = pounds of water fed per hour ¾ B. t. u. absorbed by one pound of water. B. t. u. per pound of water is made up of three parts, assumed as follows:

One B. t. u. is absorbed for each degree (F.) rise up to 212° Fahrenheit. As the water is turned into heated condition from cold, 970 B. t. u. are absorbed. For each degree of heat above 212° F. then added, up to temperature of exhaust gases, 0.48 B. t. u. is absorbed.

This method is not scientifically accurate because of pressures other than atmospheric imposed upon heat in the exhaust and its consequential vapor, however the difference is negligible in general calculation.

To Find the Pitch of Propeller in Practical Way:

A simple way to find the pitch of a propeller is as follows: At some place on the blades of every propeller there will be a place that is at an angle of 45 degrees to the center line of the shaft. Secure a 45 degree (draftsman's) triangle and find the place on one of the blades where it can be applied. Next measure the diameter of the propeller at the place where you have applied the 45 degree triangle, and, finally, multiply that diameter, measured in feet, by 3.1416 and that gives, directly, the pitch in feet. For all practical purposes, this is the average pitch of the propeller.

Suppose, for example, that you have a propeller that measures over the tips of the blades, 8 feet; also, that the place where the angle of the blade is at 45 degrees with the center line of the shaft is, we will say, at 4 feet diameter, or 2 feet from the center of the bore of the propeller, which is the same thing. Then, following the rule given above, we have:

$3.1416 \times 4 = 12.5664$ feet, pitch, average of that propeller.

How to Find the Decimal Part of a Mile that a Propeller Will Advance in One Revolution, or the Constant of a Propeller, that When Multiplied by the Number of Revolutions in a Given Time, Will Give the Distance the Propeller Has Advanced (hence also the ship) in the Same Time:

A nautical mile is said to be 6080 feet; or it may, for explanatory purposes, be stated thus:

1 mile (nautical) = 6080 feet.

Any number of feet less than 6,080 will first be stated like this, when it is desired to convert into decimals of a mile:

Example:

Find the velocity of water in feet per second, in two discharge pipes, one 4½" and the other 3"; also the difference in velocity. Diameter of water end of pump is 4 inches, stroke 12 inches, 60 strokes per minute and piston rod 1½" in diameter.

Solution:

$4 \times .7854 = 12.5664$ sq. in., area of cylinder head;

$1\frac{1}{2} \times .7854 = 1.7671$ sq. in., area of piston rod;

$12.5664 = 1.7671 - 2 = 11.6829$ sq. in. to good, per stroke;

$11.6829 \times 12 \times 60 = 8411.688$ cu. in. water delivered per min.

$4\frac{1}{2} \times .7854 = 15.9043$ sq. in., area of $4\frac{1}{2}$ in. pipe;

$8411.688 - 15.9043 \times 60 \times 12 = .734$ ft. per second;

$3 \times .7854 = 7.0686$ sq. in., area of 3″ pipe;

$8411.688 - 7.0686 \times 60 \times 12 = 1.652$ ft. per sec. in 3″ pipe;

$1.652 - .734 = .918$ ft. per second, difference in velocity.

Revolution Calculation:

At 10:00 A. M. the counter reads 956780, at 10:50 you set the clock back 10 minutes, what will the counter read at 11:00 A. M. if the engines are making 332 R. P. M.?

Solution:—From 10:00 A. M. to 11:00 A. M. is 60 minutes, to which must be added the 10 minutes the clock was set back as the engine has actually to run 70 minutes before 11:00 A. M. If the engine, or engines, are making 332 R. P. M. for 70 minutes it will be $332 \times 70 = 23240$ R. P. M.

Then: $956780 \div 23240 = 980020$ reading at 11:00 A. M.

STANDARD TESTS APPLIED TO INTERNAL COMBUSTION MACHINERY

Brake Horse Power:

The determination of brake horse power is the same for internal combustion engines as for steam driven engines.

Measurement of Heat-Units Consumed by Engine:

The number of heat units used is found by multiplying the number of pounds of oil or the cubic feet of gas consumed by the total heat of combustion of the fuel as determined by the calorimeter test.

In establishing the total heat of combustion no deduction is made for the latent heat of the water vapor in the products of combustion.

Measurement of Jacket-Water to Cylinder or Cylinders:

In measuring the jacket-water the method of passing it through a water meter, or to have it flowing from a measuring tank on its discharge, is reliable.

Indicated Horse-Power:

Accurate tests made to establish the Indicated Horse-Power must be in a manner that no possible reaction can occur to engine, by unnecessary bends in exhaust piping. All connection necessary should be, if possible, to the cylinder head. The use of Steam Indicators in connection with test should be avoided and special types employed manufactured for Internal Combustion Engines.

Standards for Economy and Efficiencies:

Comparison tests between steam and Internal Combustion Engines require actual generating processes employed for either prime mover. It is imperative to confine the actual losses incurred by either engine through its respective method of heat energy produced.

Thermal Efficiency:

In determining the thermal efficiency ratio per Indicated Horse-Power or per brake horse-power for internal combustion engines, the following formula may be used, expressed by fractions:

$$\frac{2545}{\text{B. T. U. per H. P. per hour}}$$

Dimensions:

It is recommended that following procedure be taken, in establishing accurate tests of engine: Take the dimensions of the cylinder or cylinders whether already known or not. The proper time to ascertain this is while the cylinder or cylinders are hot and in working order. In case where wear is shown, determine the average. Also measure the compression space or clearance volume, which should be done, if practicable, by filling the spaces with water previously measured, the proper correction being made for the temperature.

Fuel:

The fuel used for the test should be specified and the correct highest calorific value for fuel used known, to determine the maximum efficiency of engine.

METRIC CONVERSION TABLE
(SOLIDS AND LIQUIDS)

Millimetre × .03937 = Inches.

Millimetres × 25.4 = Inches.

Centimetres × .3937 = Inches.

Centimetres ÷ 2.54 = Inches.

Metres × 39.37 = Inches.

Metres × 3.281 = Feet.

Metres × 1.094 = Yards.

Kilometres × .621 = Miles.

Kilometres ÷ 1.6093 = Miles.

Kilometres × 3280.7 = Feet.

Square Millimetres × .0155 = Sq. Inches.

Square Millimetres ÷ 645.1 = Sq. Inches.

Square Centimetres × .155 = Sq. Inches.

Square Centimetres ÷ 6.451 = Sq. Inches.

Square Metres × 10.764 = Sq. Feet.

Square Kilometres × 247.1 = Acres.

Cubic Centimetres ÷ 16.383 = Cu. Inches.

Cubic Centimeters ÷ 3.69 = Fluid Drachms.

Cubic Centimetres ÷ 29.57 = Fluid Ounces.

Hectare × 2.471 = Acres.

Cubic Metres × 35.315 = Cubic Feet.

Cubic Metres × 1.308 = Cubic Yards.

Cubic Metres × 264.2 Gallons (231 Cu. Ins.)

Litres × 61.022 = Cubic Inches.

Litres × 33.84 = Fluid Ounces.

Litres × .2642 = Gallons (231 Cu. Ins.)

Litres ÷ 3.78 = Gallons (231 Cu. Ins.)

Litres ÷ 28.316 = Cubic Feet.

Hectolitres × 3.531 = Cubic Feet.

Hectolitres × 2. 84 = Bushels (2150.42 Cu. Ins.)

Hectolitres × .131 = Cubic Yards.

Hectolitres ÷ 26.42 = Gallons (231 Cu. Ins.)

Grammes × 15.432 = Grains.

Grammes ÷ 981 = Dynes.

Grammes (Water) ÷ 29.57 = Fluid Ounces.

Grammes ÷ 28.35 = Ounces Avoirdupois.

Grammes Per Cu. Cent. ÷ 27.7 = Lbs. Per Cu. Ins.

Joule × .7373 = Foot Pounds.

Kilogrammes × 2.2046 = Pounds.

Kilogrammes × 35.3 = Ounces Avoirdupois.

Kilogrammes ÷ 1102.3 = Tons (2000 Lbs.)

Kilogrammes Per Sq. Cent. × 14.223 = Lbs. Per Sq. Inch.

Kilogrammes Metres × 7.233 = Foot Pounds.

Kilo Per Metre × .672 = Lbs. Per Foot.

Kilo Per Cu. Metre × .026 = Lbs. Per Cu. Foot.

Kilo Per Cheval × 2.235 = Lbs. Per H. P.

Kilo-Watts × 1.34 = Horse Power.

Watts ÷ 746. = Horse Power.

Watts ÷ .7373 = Foot Lbs. Per Second.

Calorie × 3.968 = B. T. U.

Cheval Vapeur × 3.968 = Horse Power.

(Centigrade × 1.8) + 32 = Degrees Fahrenheit.

Gravity Paris = 980.94 Centimetres Per Sec.

POWER EQUIVALENTS.

One Horse Power Is Equal to:

1,980,000	foot pounds per hour
33,000	foot pounds per minute
550	foot pounds per second

273,740	---------kilogram metres per hour
4,562.3	---------kilogram metres per minute
76.04	---------kilogram metres per second
2,552	---------British Thermal Unit per hour
42.53	---------British Thermal Unit per minute
0.709	---------British Thermal Unit per second
0.746	---------Kilowatt
746	---------Watts

One Kilowatt Is Equal to:

2,654,400	---------foot pounds per hour
44,239	---------foot pounds per minute
737.3	---------foot pounds per second
366,970	---------kilogram metres per hour
6,116.2	---------kilogram metres per minute
101.94	---------kilogram metres per second
3,438.4	---------British Thermal Unit per hour
57.30	---------British Thermal Unit per minute
0.955	---------British Thermal Unit per minute
1,000	---------Watts
1.34	---------horse power

One Watt Is Equal to:

2,654.4	---------foot pounds per hour
44.239	---------foot pounds per minute
0.737	---------foot pounds per second
366.97	---------kilogram meters per hour
6.12	---------kilogram metres per minute
0.102	---------kilogram metres per second
3.4384	---------British Thermal Unit per hour
0.0573	---------British Thermal Unit per minute
0.000955	---------British Thermal Unit per second
0.001	---------Kilowatt
0.001.340.6	---------Horse power

One Foot Pound Is Equal to:

0.0000003767	---------Kilowatt per hour
0.0000226	---------Kilowatt per minute
0.001356	---------Kilowatt per second
0.000000506	---------Horse Power per hour
0.0000303	---------Horse Power per minute
0.001818	---------Horse Power per second
0.0003767	---------Watt per hour
0.0226	---------Watt per minute
1.356	---------Watt per second

One Foot Pound Is Equal to:

1.3825	---------Kilogram metres
0.001288	---------British Thermal Unit

CHAPTER IV.

PRINCIPLES OF DIESEL OPERATION

It must be admitted that great progress has been made of late to standardize Diesel machinery on same basis as found to day among steam driven engines. In fact, a similar plan will be adopted as time advances. It is true, that mechanical contrivances covered by numerous patents are held in many instances the most vital part in operation. It is also true, that the same condition existed a number of years ago when steam machinery was in its pioneer days.

By careful observation in pratical performance engineers were enabled to find methods of better results in steam engineering. Even the most insignificant defects were found to be worthy of consideration. The result was the standardization of reciprocating steam machinery. So it is to-day a fact, that in this respect a standard in construction was made possible and very little difference exists in this type of machine.

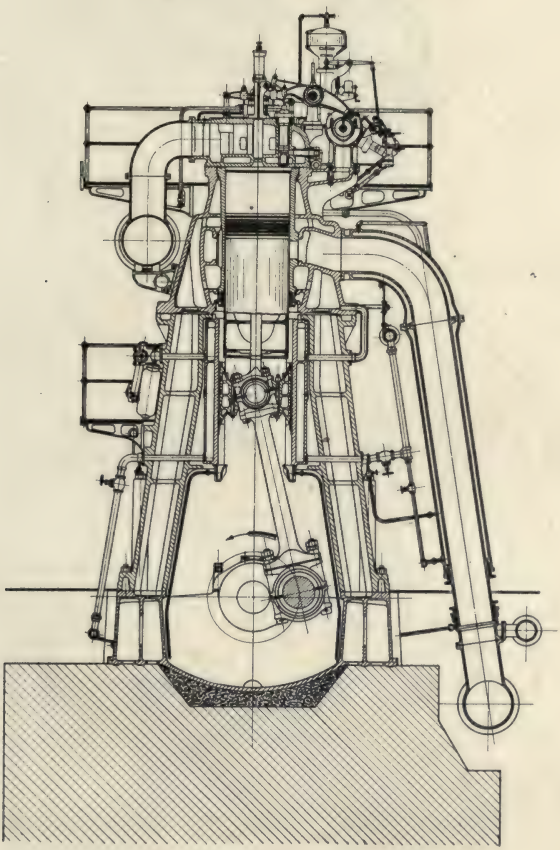
The same was accomplished with the Internal Explosion Engine, or such machinery receiving their power stroke by impulse of explosion such as the gasoline driven engine. As will be observed, the prevailing principle is identical in every respect. In either the two-stroke cycle or engines following the principle of four-stroke cycle the relative identity in construction has been accomplished. While manufacturers in some instant adhering to the overhead-valve, T-head cylinder or L-head type, etc., nevertheless there is a prevailing standard governing the system as a whole. It will be acknowledged, that in this type of engine construction a great deal of improvement will be accomplished, but the laws established will remain.

This, of course, would be impossible in the case of Diesels. Problems of reversing of Diesel engines and fuel injection processes might easily find solution. Valve arrangements necessary to reverse the power plant depends a great deal on future development. The usual procedure in accomplishing this is by cam operation. In most cases there are levers by which cams are operated causing the opening of its respective ports.

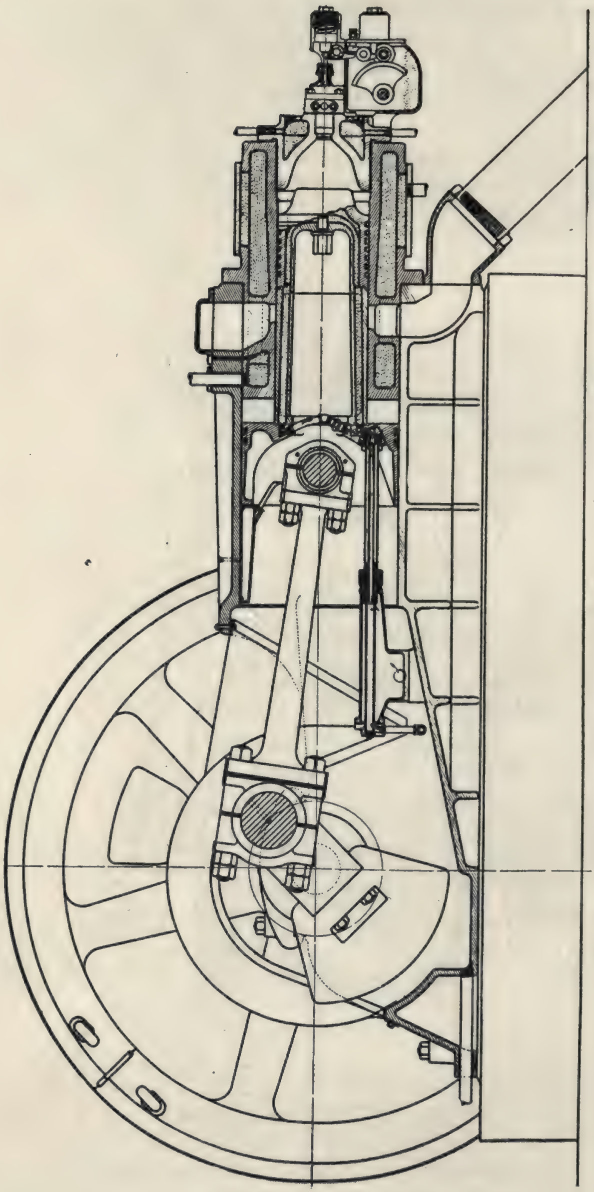
In most engines of marine and stationary types, horizontal cam-action is extensively employed. In this case, where two sets of cams performing the purpose of the alteration, cams are generally directly in conjunction with shaft. Opening of valves is thereby accomplished in the movement of the shaft in longitudinal direction. In some cases independent cam arrangements are favored by builders.

The methods of fuel injection is a subject which can be solved and ultimately will have to be considered. Arguments in favor of Solid Injection as opposed to Air Injection is merely a matter of opinion. Both systems have proven satisfactory and the fuel consumption may be considered nearly on an equal. The principal reason advanced against the use of Air Injection may be expressed through the necessity of compressor equipment in the case of the latter system.

Compressor installation is imperative. Even with the employment of solid injection provision must be made to supply air for starting purposes. There are commendable features in the use of solid injection, but not important enough to make this method exclusive in adoption.



Cross-sectional view of Nordberg Diesel Engine. In this type, similar to the Busch-Sulzer Engine, a true representation of two-cycle principle of construction is to be found.



Longitudinal view through Standard horizontal type of two-cycle Diesel engine. In mechanical simplicity this type is noticeable.

In the comparison between the two-cycle and four cycle type the primary difference will be found in the arrangement of valves and the requirement of the scavenging pump on the two-cycle engine.

Added mechanism, such as valves, etc., necessitates added expenses on the four-cycle than on two-cycle construction. These again are matters which after carefully going in detail are not seriously acting against this type of construction.

The introduction of two-cycle double acting engines appears not to meet with favor in the United States. There is no question as to the advantages of this class of machine. It may well be stated that the two-cycle double acting engine is a machine of high merits owing to its enormous developing of power. In economy it has no equal. It differs inasmuch as each stroke is a working stroke. The distribution of power in its general arrangement of cylinders gives an exertion of double action through construction of seperate cylinders receiving its impulse through seperate inlet valves above and below.

The reason for lack of interest in this type of construction in the United States, may be found in the fact that a tendency exists in the adoption of large types of engines with the simplest mechanism. This theory is indisputably the best. An engine built to perform work should be constructed of good material and all necessary complicated mechanism eliminated.

The cycle of operation governing Diesels is exceedingly simple, being based on two natural laws—

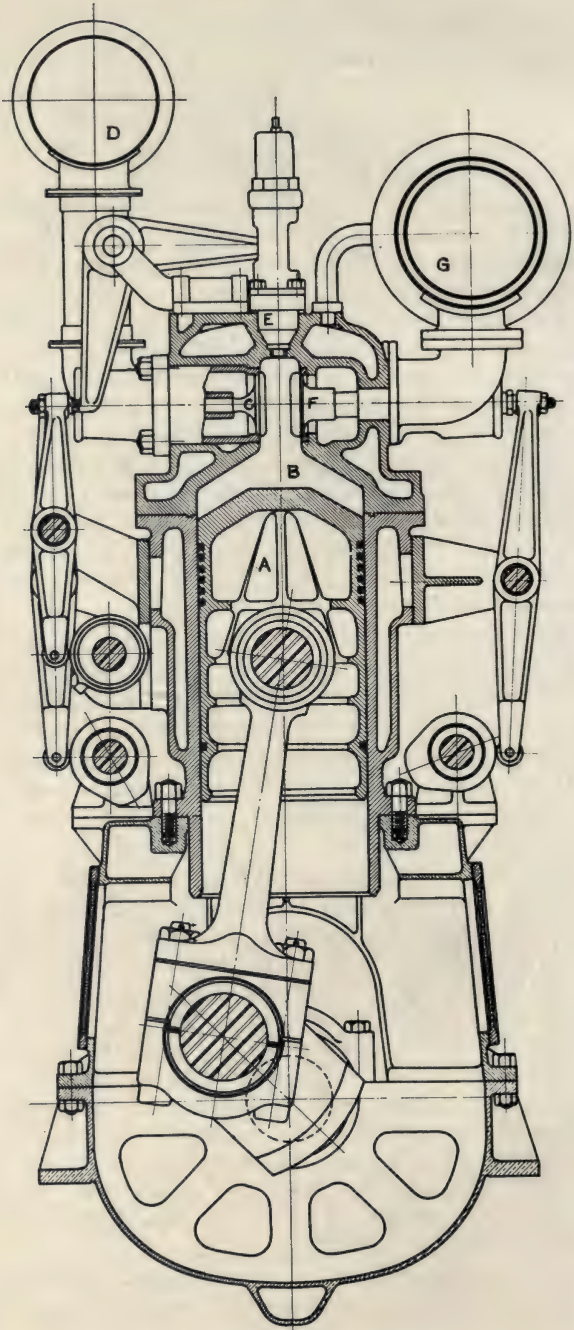
First—Air compressed in a closed cylinder will develop heat in proportion to the degree and character of compression.

Second—Any product of petroleum designated in trade as crude or fuels oils, if properly atomized, will ignite spontaneously when injected in a cylinder of compressed air, whose temperature has been raised above the fire test of the fuel.

The Diesel engine differs from a gas engine from the fact that it is a **constant pressure** engine. That is, the injection of fuel into the cylinder is timed and controlled to maintain constant pressure during its introduction, while in a gas engine a constant volume of mixture is taken into the cylinder and after compressing same to a safe limit (about 70 lbs.) the mixture is ignited by some auxiliary mechanism and the pressure instantly rises in the nature of an explosion to a degree depending on the volume of the mixture.

Now while the terminal pressures are about the same in both types of engines, the efficiency of the Diesel is twice that of the gas engine due to compressing a non-explosion fluid to a high temperature before injecting the fuel, then controlling this injection so that combustion continues for a pre-determined time, varying with the load, before expansion begins.

This process enables the Diesel to maintain a much higher mean effective pressure with a corresponding greater horse power for the same cylinder dimensions.



Four-cycle Type (Nelseco).

In addition to the high thermal efficiency of the Diesel engine, a great commercial advantage is obtained from the fact that the cycle or method of ignition permits the use of a cheap, low-grade fuels of high heat value; and it has been demonstrated that all petroleums products, either crude or refined, obtained anywhere, can be burned with high economy and certainty in the same engine, by merely changing some minor adjustment of the fuel injection mechanism.

The designation of the cycle implies that the work in the cylinder is accomplished in two-cycle engine in two strokes, or a power stroke every full revolution and in the four-cycle engine the work is accomplished in four strokes of the piston, or two revolutions of the engine.

When defining the action of power development on a four-cycle engine following procedure occurs: The first downward stroke draws into the cylinder pure air only at atmospheric pressure and temperature.

The second stroke compresses this air to about 450 to 600 lbs., per square inch and increases its temperature to about from 1000° Fahrenheit up to 1150° F. depending upon the design.

On the third or second downward stroke, a pre-determined amount of fuel, which has been delivered to the fuel valve by the pumps on the air admission stroke of the engine, is forced into the cylinder by means of air compressed to a higher pressure than that of the air in the cylinders. The fuel is ignited by the heat of the compressed cylinder charge and by expansion drives the piston downward.

The fourth or second upward stroke drives the spent gases out of the cylinders.

It will be noted that while the sequence of the strokes forming this cycle is similar to that of the gas engine, the functions performed during the cycle (except on the exhaust stroke) are entirely different.

On the first stroke, by taking into the cylinder pure air only, we are enabled to compress this charge sufficiently to secure a very high temperature, which would be impossible with a charge of gas mixture, because of liability to pre-ignition.

This high temperature secured allows the use of cheap, low-grade fuels of high fire test, without risk of explosion, and increases both the thermal and commercial efficiency of the engine.

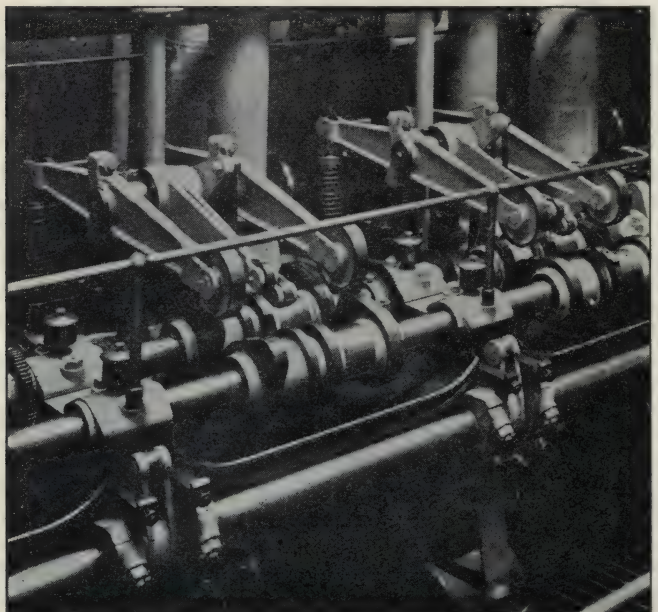
On the third stroke the fuel can be introduced under absolute control as to timing and quantity, maintaining a constant pressure in the cylinder for a period depending on the work required from the cylinder at that moment. The use of compressed air for injection, where this system is used, thoroughly atomizes the fuel and prepares it for instant firing as introduced.

As stated above, this combustion is not in any sense an explosion, but takes place during a well-defined, pre-determined portion of the power stroke and at constant pressure, and because of the nature of its introduction into the cylinder and the large volume of pure air into which it is forced, the combustion is perfect within the range of cylinder power rating, hence the high thermal efficiency of the Diesel cycle on low grade fuels.

The foregoing description of the engine and its cycle is, of course, general in character—intended to cover the principles around which the physical construction is assembled.

The method of starting by compressed air is to cause air pressure through the medium of independent compressors on larger engines or in some instances direct connected compressors in smaller types, to be exerted through the assistance of a starting valve on the pistons.

The process in starting engines differs somewhat, depending upon the general construction. The usual proceeding is to bring the starting lever in its starting position. In doing so, the lever actuating the fuel valve is brought in a non-operating position. In this state the fuel valve remains closed in its first operation.



Uniform operation depends greatly on accurate functioning of valve-arrangement actuated by cams.

The starting valve in performing its function when opened, to admit air to either cylinder, the engine begins to turn. When the piston passes its dead center a charge of oil is allowed to enter into the cylinder causing a volume of gas to be established in the combustion chamber.

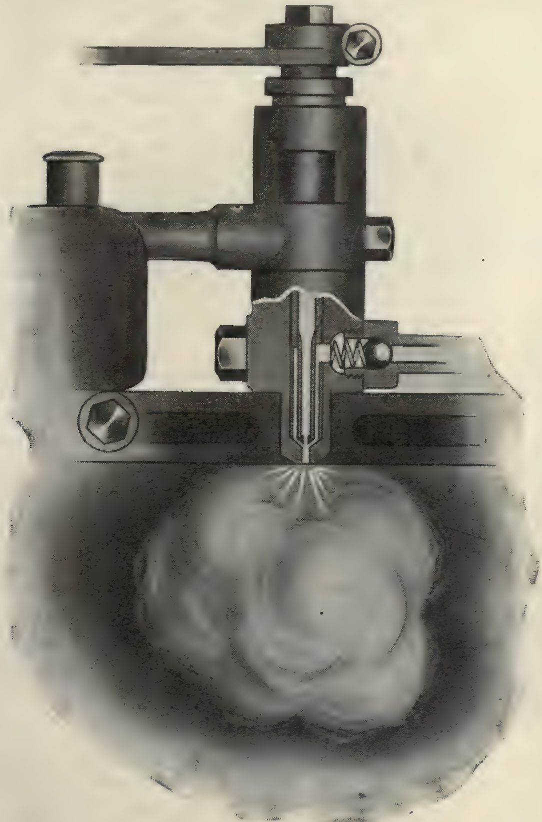


Illustration demonstrating "Interior Action" of Fuel being brought in contact with heat temperature.

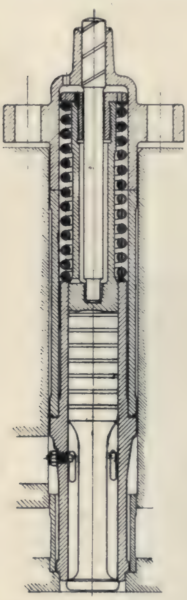
It is advisable to allow the engine to turn over several revolutions by air to cause a uniform heating to be created. This has the effect of causing an almost immediate combustion to take place when the starting handle is brought in the respective position, allowing the fuel valve to operate by its own mechanism.

The starting reservoir having for its object the storing of air for starting purposes should be filled to its full capacity before starting of engine. The fuel valve, which is pumped up by hand before

commencing the operation of the engine, and its pipe connections should be properly filled with oil.

At all times make sure that a full supply of air is on hand for the operation of the fuel valve.

Proper operation of engine will be the result when following instructions are carefully considered: (1) Examine fuel tanks; (2) See that the lubrication system is properly functioning; (3) Test the oil pump; (4) Test the water pump; (5) See that all connections are tight; (6) See that the compressor is in proper working condition; (7) Watch all gauges; (8) See that seacock is open (on marine work); (9) Examine all piping; (10) Go carefully over all parts of the mechanism and examine all nuts; (11) Try all levers; (12) If on marine work, make sure that the Annunciator is in proper operation.



Starting Valve of Carels type used on Nordberg Diesels.

The functions of the cams may be explained in brief as mechanical arrangements, having for their object the regulation of exact time required in opening of valves, relative to the position of the piston, causing a regular functioning in general operation.

In the action of the four-cycle engine it is understood that each valve is required to open in two revolutions, it follows then that the cam shaft must rotate at half the speed of the crankshaft.

The cam operating the fuel valve performs its function in corresponding period to the piston stroke, depending upon established condition. The exhaust valve cam in its relation to the working stroke, allows the valve to stay open during the entire stroke and again allows the exhaust valve to close after the top dead center has been reached.

The admission of air is accomplished by the valve being kept open admitting the air during the downward stroke and again closes after the crank passes the bottom dead center.

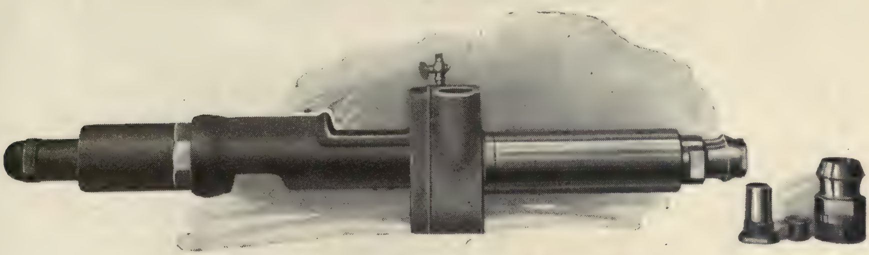
The action of the starting valve cam causes the valve to open before the top dead center has been reached and closing the same before the end of the stroke.

INJECTION OF FUEL.

It will be noted, when studying the different methods of fuel-injection, that considerable difference in design are to be found. While the trend appears to be towards the solid injection system, nevertheless, mechanical injection is prevalent. It must be acknowledged that there are some advantages in the use of solid injection. In particular the elimination of air as the primary factor in forcing the oil into its

receptacle. It is true, that the compressor is a part of the engine room equipment, imperative as an auxiliary machine. A Diesel plant without the provisioncy of air for starting purposes must be dismissed.

Definition of Fuel Valves: Generally speaking, there are two types of valves employed in the various Diesel engines, namely the closed nozzle and the open nozzle valves. While on vertical types of Diesels the closed nozzle is principally used, owing to the structural reasons, the open nozzle is prevalent on horizontal engines.



In the "Open Nozzle" Spray Valve, as adopted by the Snow Oil Engines, the charges are consumed with accurate delivery, irrespective of load-variation.

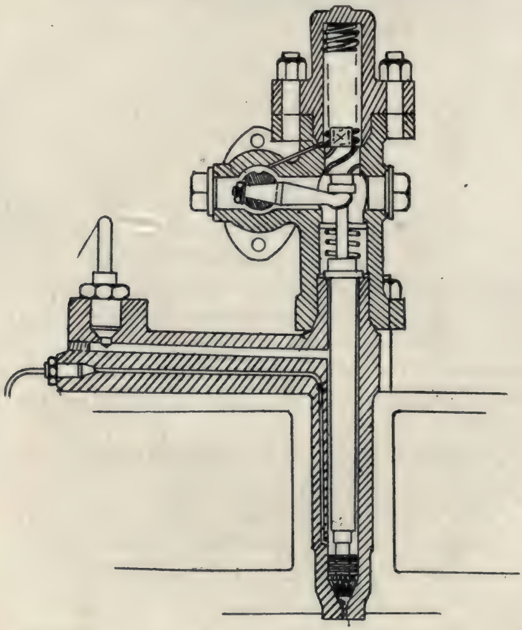
Open Nozzle Fuel Valve: This particular type of valve, or properly speaking, nozzle, is designed to act as a receptacle wherein a needle valve controls the flow of air to the atomizer tip. This needle valve automatically opens to allow the uniform distribution of oil to exist, depending in its operation by proper actuation of either a cam device or in some types on rocker arm arrangement. A small cavity is interposed between the valve and the cylinder, in some cases an enlargement of the passage to the cylinder, allowing the fuel to be deposited.

Inasmuch, as the fuel pump depends in its entirety on the proper governing to conform with the desired quantity necessary to keep the engine in regularity supplied, the needle valve performs its function of opening and closing at regular intervals. When opening this valve allows the air at the proper time from the compressor to enter, sweeping the oil charge along and carrying it into the cylinder. As the oil enters the extreme end of the nozzle, it is swirled by the force of air through a set of perforated disks, serving to break up the oil into particles while entering the combustion chamber.

Interior Action of Fuel Coming in Contact With High Temperatures: The existing high temperature in the combustion chamber of a normal value corresponding to 550 lbs. compression pressure should be at least 1000 degree Fahrenheit when the engine is on its commencement. The usual temperature on a well designed engine when in proper working order and during operation may be well above 1400

degrees Fahrenheit. Modern engines are well protected against leakages, and rarely any loss of efficiency is due to this defect so frequent on older types. With the proper timing of the fuel injection valve very little trouble will be experienced, it should be realized, that the change of oil, varying in specific gravities, necessitates a careful observation and often requires the re-timing of valves. Again trouble may be experienced by "air-pockets" in the fuel oil, causing the lack of proper flow, which incidentally causes serious neglect in proper functioning of oil distribution into the cylinder. Water in oil causes carbonizing and dangerously effects the efficiency of cylinder performances.

Closed Nozzle Fuel Valve: This type has been used on earlier Diesels. The advantages in employing the closed valve is, if it may be considered an advantage, that it deposits the oil in a receptacle entirely isolated from the influence of the hot compressed air in the cylinder. While in this construction the oil in reality enters the cylinder ahead

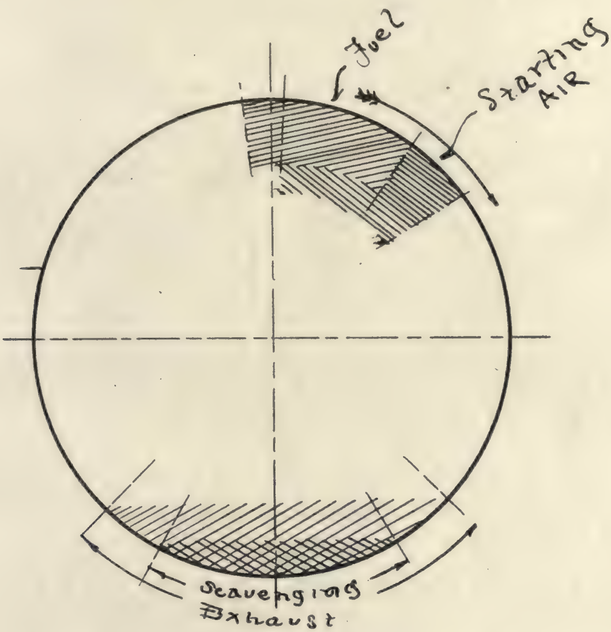


Cross-sectional view of Busch-Sulzer characteristic Fuel-Injection System. Note air-distribution.

of the air, ignites and very often endangers the efficiency of cylinder performances by entering without being thoroughly atomized. This is principally due on account of the initial charge entering under somewhat lower pressure, preventing a thorough breaking up of the fuel oil. To overcome this detrimental defective existency, the employment of

higher air pressure becomes imperative. The fuel valve, differing but little from the open-nozzle type, with the exception that the needle valve is located below the cavity in which the atomizing takes place and is, in similarity to the open-nozzle valve in direct connection with the air line. Owing to its construction, by which the fuel oil pump direct delivers the oil towards the oil chamber, the pressure necessary in the air line is usually no less than 900 lbs., per square inch. When the compression pressure is in the cylinder somewhat around from 500 to 550 lbs., per square inch the needle valve opens and allows the charge to enter. All other performances, such as the breaking up of the oil are similar to the open-nozzle type.

Timing of Valves: Timing of valves is not very difficult as some novices on Diesel machinery are apt to believe. We will first take up the two-stroke cycle, which by nature of construction is the simplest engine.



Valve Settings of Simple Port Scavenging Two-Cycle Engine.

Timing Two-cycle Engine: The two-cycle engine exerts a "power-stroke" every revolution. Unlike the four-cycle type, using valves in its entirety, the two-cycle engine eliminates the air inlet stroke and the exhaust stroke and employs ports near the bottom of the cylinder liner, through which the burnt gases are driven by a current of air.

The entering of the air into the cylinders differs in the various types of two-cycles. In some engines it enters by valves in the cylinder head and others again by ports similar and opposite the exhaust ports, and covered and uncovered by the piston. The fuel valve opens in most cases at 5 degrees before the top center and closes at 42 degrees over the top center. Expansion occurs until the crank reaches a point about 40 degrees from the bottom center. At this period the piston uncovers the exhaust ports near the bottom of the cylinder, allowing the products of combustion to escape. About 10 degrees later the air inlet valves, or scavenging valves, open, through which air is blown into the cylinder, cleaning out the remaining burnt gases, and in consequence leaving the cylinder full of pure air. At 40 degrees over the bottom center the exhaust ports are closed by the piston on its upward stroke, and about 20 degrees later the scavenging valves close. At this period compression of air begins, receiving the fuel at 5 degrees before top center.

TIMING DIAGRAMS

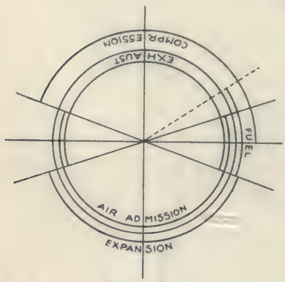


Fig. A.

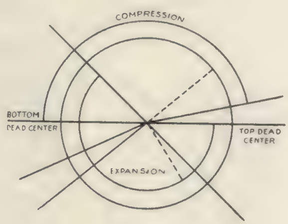


Fig. B.

It will be noted that in Figure A, pertaining to timing of four-cycle engine, that the fuel admission (a part of the cycle performance), corresponds with the actual requirement of engine performances, varying in load capacities.

On the two-cycle diagram, Figure B, the timing must correspond with features demanded in two-cycle operation, carrying with it scavenging performances, again depending mainly on load variations.

Timing Four-cycle Engine: While it is claimed that the four-cycle Diesel engine operates on the Otto cycle, nevertheless the cycle performances of the Diesel engine may well be considered a peculiar and most distinctive exclusive cycle of its own. The Diesel engine is a "constant pressure" engine and entirely separate in this respect from any other prime-mover. We will go over the actual performances of the four-cycle engine and follow its operation: All internal combustion engines depending for their maintenance upon four basic principles of performances, namely, (1) Admission, (2) Compression, (3) Power or Working Stroke, and (4) Exhaust. It is true, that there are two more when scavenging performances are to be considered.

And it is correct to add to those four basic principles, in particular on two-cycle performances (5) Air intake for scavenging, and (6) scavenging performances. But we are primarily interested in the four standard principles of operation. Let us carefully consider each performance.

First: On the first downward stroke of the ycle air is drawn from the outside source into the cylinder through the air-inlet valve.

Second: On its upward stroke the air being compressed to the universally known figure of about 500 pounds per square inch. This compression pressure raises the temperature of the air to about 1000 degrees Fahrenheit.

Shortly before the end of the stroke, fuel oil is injected into the place, known as the combustion chamber. This place is identical with the clearance space of a steam cylinder, namely, the space between the piston face and the cylinder head. Piston (on the vertical) being on its upward stroke, and on the horizontal on the outward stroke. This oil, now coming in direct contact with the existing high temperature, begins to burn—causing the combustible substance which follows the laws of least resistance, which, being exerted against the piston, causes the reciprocation with the consequential results of revolution of the engine.

Third: As previously explained, the number three stroke is the most important, the power or working stroke. The fuel on the commencement of this stroke is cut off at about 1/10 of the stroke, combustion having taken place, with the expansion, which follows near the end of the stroke, the exhaust valve opens.

Fourth: This stroke merely causes the burned gases to be expelled, after which the cycle of operations is repeated.

We will now follow the actual work taking place during the power generation. Inasmuch as the four-cycle engine gives two revolutions of the crank, or four strokes to a cycle, the power exertion of four-cycle engines is a power stroke every second revolution. In most engines the air inlet valve opens 20 degrees before the top center, and closes 16 degrees after bottom center, giving a total angular opening of 216 degrees of the crank. Immediately after the air inlet valve is closed, the air is compressed until a point about 5 degrees before the top center is reached. Admission of fuel now commences, continuing until the crank is 40 degrees over top center, giving the fuel valve an angular opening of 45 degrees.

As the gases expand, forcing the piston down until 34 degrees before the bottom center, the exhaust valve opens and the products of combustion are released and then expelled by the fourth stroke, or the upward stroke, of the piston. The exhaust valve closing 11 degrees after the top center.

Mechanical Timing of Valves: To accurately ascertain the mechanical timing arrangement of valves, it is best to time each cylinder in rotation. Begin to time the exhaust and admission valve on front cylinder and, when all marks have been properly made, corresponding

to the flywheel, follow the exact measurement on each corresponding cylinder.

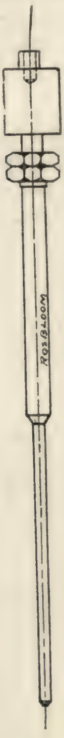
To establish the flywheel position, bringing it in the proportional requirement corresponding to upper dead center of the crank should be the first step. The valve cages can then be removed, and the distance from the surface of the cylinder head to the piston, establishing the clearance, can be properly determined. When this has been established mark flywheel. If trammel is used, be careful in finding the retangular advance of the crank on its upward stroke, by which the numerical opening and closing of each valve gives an accurate idea. A steel tape may be used to bring the opening of exhaust valve to a point of correctness. Since the timing given is in degrees, the value must be transformed into inches on the flywheel circle.

The exhaust cam rocker must be firmly brought in contact both with the cam and with the valve stem. After the mark on the top dead center of crank has been thoroughly established, turn engine over at least 12 degrees to make positive that the correct seating of valve has been accomplished. If undue valve checking is experienced, the operator should carefully examine the setting. If the irregularity does not exceed a few inches on trammel mark on flywheel, the clearance between the valve rocker and the cam may be adjusted bringing the setting back to the stated values. To the man inexperienced it is best to allow a little clearance, adjusting the cam on earlier or later cut off, after being thoroughly convinced that the exhaust charges show an excessive smoke. At all times follow the exact routine in following manner: The exhaust opening of number 1 will be set; the admission closing of a second cylinder will be checked. It will be noted, that in case any irregularity exists, the fault may be detected by a peculiar pounding in the cylinder after engine is in motion. On marine engines, valves should be properly adjusted and all care taken that the tightness of the same are accomplished.

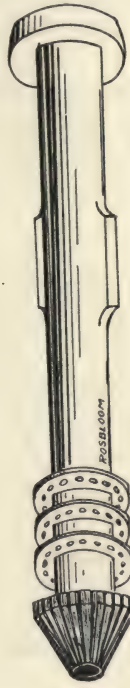
It is necessary in operation of Diesels that the injection air pressure be altered to conform to load changes. It is clear that with load changes the time during which the fuel is injected should also vary. On low loads the amount of fuel oil is small and will be entirely blown into the cylinder long before the valve closes. The balance of this period of valve opening is taking up with the injection-high-pressure air. All these calculations are imperative in successful operation and requires careful observation.

Cleaning Valves: To maintain efficiency and avoidance of breakdowns, the inspection of valves must be made a matter of routine work. At least every month each valve and cage should be given a thorough overhauling. Spare parts should be at hand and a reserve set of valves should never be neglected. When the old valves are taken off the engine and substituted by a reserve valve, it should be taken apart and inspected. Keeping foreign matter from settling around the seats and springs by giving them a bath in gasoline is imperative.

Occasional grinding of valves becomes necessary. In particular in cases where the fuel contains high percentages of asphalt or sulphur. Corrosion and pitting are primary causes of leakage. Chemical action of the exhaust gases rapidly deterriorates the material with increased faulty action of the entire unit. Every part, no matter how insignificant is in harmony with the plant, and the neglect of even the smallest parts of the engine, may cause serious breakdowns.



Valve Spindle.



Sprayer.

Timing of Fuel Valves: The timing of fuel valves depends a great deal on the particular make. While different designs are forthcoming as Diesel machinery progresses, nevertheless the common procedure is as follows: Turn engine over until it passes the desired mark of fuel valve opening. The air line valve is "cracked," at about 75 pounds air pressure on the fuel valve. The engine now being turned over slowly until the trammel cuts the mark on the flywheel signifying opening. At this time the injection valve should start to open, which may be evidenced by the sound of injection air blowing into the cylinder. The decrease or increase of valve opening, may be adjusted on the rocker arm clearance, depending upon the desired position of the valve arrangement. When the engine is brought to its proper position corresponding

to fuel valve adjustment, it should be noticed by the ceasing of escaping air. To bring the adjustment in accurate desired valve operation, it will be found necessary to turn the engine back after the point of "cut-off" has been determined. Bring the engine back ahead of the valve opening mark and shift the cam nose to produce the required opening with the roller clearance in correct position. To ascertain the correct setting after the checking has been performed, it is recommended to give the engine several turns. It will be noticed that the difference in adjustment of either early or late may be caused by shifting the nose back or forth as the case may be, which will have an effect on the roller clearance, tending to bring the valve to its desired correctness.

The equipping of Fuel Valve Timing Control, adopted by Standard firms, such as the Busch-Sulzer, Nordberg-Manufacturing Company, etc. by which the alteration of injections automatically takes place, eliminates a great deal of complications found on the usual types.

TENSILE STRENGTHS OF MATERIALS

Cast and Rolled Metals.

Pounds per Square Inch.

Aluminum, Cast		15,000
Aluminum, Bars		28,000
Brass, Cast, Yellow		27,000
Brass, Rod		55,000
Brass, Rolled, Naval	1" and below	62,000
	Above 1" to 2½"	60,000
Bronze, Cast, Steam		30.000-36,000
Bronze, Cast, Manganese		65,000
Bronze, Rolled Manganese	1" and below	72,000
	Above 1"	70,000
Bronze, Cast, Phosphor		30,000-40,000
Bronze, Rolled, Phosphor	½" and below	80,000
	Above ½" to 1"	60,000
	Above 1"	55,000
Iron, Cast, Grey	Light castings	18,000
	Medium castings	21,000
	Heavy castings	24,000
Iron, Malleable		40,000
Iron, Wrought Shapes		48,000

Lead, Cast		1,600- 2,400
Monel, Cast		65,000
Monel, Rolled	1″ and below	84,000
	Above 1″ to 2½″	80,000
	Above 2½″	75,000
Nickel, Cast		85,000
Nickel, Rolled		96,000
Steel, Cast	Hard	80,000
	Medium	70,000
	Soft	60,000
Steel Forgings		75,000-90,000
Steel, 3.5% Nickel		100,000-105,000
Tin, Cast		4,000-5,000
Zinc, Cast		4,000- 6,000

STRENGTH OF MATERIALS

(Stresses in Thousands of Pounds)

Metals and Alloys	Tension Ultimate	Elastic Limit	Compression Ultimate	Modulus of Elasticity Pounds
Aluminum, cast	15	6.5	12	11,000,000
Copper, cast	25	6.0	40	10,000,000
Brass, 17% Zn.	32.6	8.2	42	---------
Brass, cast, common	18-24	6.0	30	9,000,000
Bronze, 8% Sn.	28.5	19.0	42	10,000,000
Lead, cast	1.8	---	--	1,000,000
Tin, cast	3.5-4.6	1.5-1.8	6	4,000,000
Zinc, cast	4-6	4	18	13,000,000
Steel, cast, soft	60	27	tensile	29,000,000
Steel, cast, medium	70	31.5	tensile	29,000,000
Steel, cast, hard	80	36	tensile	29,000,000
Cast Iron, common	15-18	6	80	12,000,000
Nickel Steel, plate	85-100	50	tensile	29,000,000
Wrought Iron, plate	48	26	tensile	28,000,000
Gold, cast	20	4	------	8,000,000
Silver, cast	40	--	------	---------

EXPANSION OF PIPES

The linear expansion and contraction of pipe, due to differences of temperature of the fluid carried and the surrounding air, must be cared for by suitable expansion joints or bends.

In order to determine the amount of expansion or contraction in a pipe line, following table demonstrates the increase in length of a pipe 100 feet long at various temperatures.

The expansion for any length of pipe may be found by taking the diference in increased length at the minimum and maximum temperatures, dividing by 100 and multiplying by the length of the line under consideration.

Expansion of Pipe

Increase in Length—Inches per 100 Feet.

Temperatures, Degrees F.	Steel	Wrought Iron	Cast Iron	Brass and Copper
0	0	0	0	0
20	.15	.15	.10	.25
40	.30	.30	.26	.45
60	.45	.45	.40	.65
80	.60	.60	.55	.90
100	.75	.80	.70	1.15
120	.90	.95	.85	1.40
140	1.10	1.15	1.00	1.65
160	1.25	1.35	1.15	1.90
180	1.45	1.50	1.30	2.15
200	1.60	1.65	1.50	2.40
220	1.80	1.85	1.65	2.65
240	2.00	2.05	1.80	2.90
260	2.15	2.20	1.95	3.15
280	2.35	2.40	2.15	3.45
300	2.50	2.60	2.35	3.75
320	2.70	2.80	2.50	4.05
340	2.90	3.05	2.70	4.35
360	3.05	3.25	2.90	4.65
380	3.25	3.45	3.10	4.95
400	3.45	3.65	3.30	5.25
420	3.70	3.90	3.50	5.60
440	3.95	4.20	3.75	5.95
460	4.20	4.45	4.00	6.30
480	4.45	4.70	4.25	6.65
500	4.70	4.90	4.45	7.05
520	4.95	5.15	4.70	7.45
540	5.20	5.40	4.95	7.85
560	5.45	5.70	5.20	8.25
580	5.70	6.00	5.45	8.65
600	6.00	6.25	5.70	9.05

Temperatures, Degrees F.	Steel	Wrought Iron	Cast Iron	Brass and Copper
620	6.30	6.55	5.95	9.50
640	6.55	6.85	6.25	9.95
660	6.90	7.20	6.55	10.40
680	7.20	7.50	6.85	10.95
700	7.50	7.85	7.15	11.40
720	7.80	8.20	7.45	11.90
740	8.20	8.55	7.80	12.40
760	8.55	8.90	8.15	12.95
780	8.95	9.30	8.50	13.50
800	9.30	9.75	8.90	14.10

CHAPTER V.

LIQUID SUBSTANCES

SPECIFIC HEAT

Bodies vary greatly in the capacity which they possess for absorbing heat under equal changes in temperature. The relation which thus exists between them is expressed by the "specific heat," which may be defined as the quantity of heat necessary to be imparted to a given body in order to raise its temperature one degree relatively to the quantity that is required to raise through one degree an equal weight of water from its point of greater density at 39.1°. Thus, for instance, one pound of air at constant pressure may be raised through one degree by the expenditure of only 0.2375 of the heat necessary to raise one pound of water through one degree; or, what amounts to the same, the amount of heat expended to raise the temperature of one pound of water by one degree would heat $1/0.2375 = 4.2105$ pounds of air through the same increment.

As the specific heat of water is greater than that of any other known substance, the specific heat of all other substances must of necessity be expressed in decimals.

Water does not absorb heat exactly in proportion to its increase in temperature; in other words, the specific heat of water varies with the temperature, as is rendered evident in the following table.

SPECIFIC GRAVITY

Water is universally adopted as the standard by which the relative weight of all liquids and solids are determined, this relation being expressed by the term of "specific gravity." The specific gravity of a body, therefore, indicates its weight as compared with that of an equal body in the form of volume of pure water, determinations of specific gravity are generally referred to the weight of one cubic foot of water at 62 degrees Fahrenheit. At the more important temperatures the weight is as follows:

Weight of one cubic foot of pure water:

At 32	degrees	F.	(freezing point)	62.418 lbs.
" 39.1	"	"	(maximum temperature)	62.425 "
" 62	"	"	(standard temperature)	62.355 "
" 212	"	"	(boiling point under atmospheric pressure	59.760 "

For general purposes the weight of water is taken in round numbers as 62.5 pounds per cubic foot. In bulk, water is usually measured by the gallon, the volume of which is 231 cubic inches (the British gallon contains 277.274 cubic inches), or 0.134 cubic feet. A gallon of water at 62 degrees, therefore, weighs slightly over 8 and 1/3 pounds, and 7.48 gallons equal one cubic foot.

PRESSURE OF WATER

From the weight of water at the standard temperature of 62 degrees, its pressure upon any exposed surface may be readily determined for any given depth or head. The weight of one cubic foot at the above temperature being 62.355 lbs., it is evident that for a head of one foot the pressure must be 62.355 lbs. per square foot, and $\frac{62.355}{144} = 0.433$ lbs. per square inch; and, further, that a pressure of one pound per square inch will be produced by a head of $\frac{1}{0.433} = 2.309$ feet.

OIL MEASUREMENTS

For the calculation of evaporative results, fuel used for power, heat value, etc., units of weights are employed, i. e., the pound, kilo, etc. (1 kilo = 2.204 pounds avoirdupois); but for measurements of bunkers, cargo tanks and in general sales contracts, the oil is figured by volume. Thus, in the United States, the usual units are the United States gallon (231 cubic inches = 3.785 litres) and the barrel of 42 gallons.

The litre and the Imperial gallon (4.54 litres) are used abroad, where the barrel is figured at 41 Imperial gallons (50 United States gallons). The Imperial gallon equals about 1.2 United States gallons. Even some of American oil firms favor the Imperial barrel (50 United States gallons) as a matter of convenience, but 42 gallons is the accepted standard.

For statistical purposes, the ton (2,240 pounds) and the "metric ton" or 1,000 kilos (2,204 pounds) are frequently used.

Volumetric measurement of oil should always be based on a standard temperature, the usual figures in this country being 62 degrees Fahrenheit. The United States Navy Department specifies 60 degrees Fahrenheit and a correction of 0.4 of 1% is made for each degree variation from this standard.

MECHANICAL EQUIVALENT OF HEAT

The mechanical unit of work is the "foot-pound," or the work required to raise one pound through the distance of one foot. The me-

chanical theory of heat regards heat as a mode of motion, an investigation has shown that there exists a definite relation between these two forms of energy, which is known as the "mechanical equivalent" of heat. That is, if, as in the experiments of Joule, a certain known amount of mechanical energy is expended (as by falling of a weight) to operate paddles in a vessel of water, the increase in temperature of the water, due to agitation by the paddles, will always be found to be proportional to the work done.

This relation or proportion is universally expressed by the amount of work necessary to raise the temperature of one pound of water through one degree Fahrenheit. The latest experimental determinations of Rowland show it to be practically 778 foot-pounds.

HEAT OF COMBUSTION:

As determined by the most recent and refined calorimetric tests, the heat of combustion, as measured by the number of B. T. u.'s that are given out upon the combustion of one pound of a given substance, is for each of the following:

Carbon burned to CO^2	14,650 B. T. U.
Carbon burned to CO	4,400 B. T. U.
Hydrogen burned to CO	62,100 B. T. U.
Marsh Gas burned to CO	23,513 B. T. U.
Olefiant Gas burned to CO	21,343 B. T. U.
Carbonic Oxide burned to CO^2	4,393 B. T. U.

BEAUME HYDROMETER SCALES

Various arbitrary scales of equal parts have been proposed for hydrometers. Of these scales those of Beaume are most extensively used. Beaumes scale for heavy liquids is constructed by locating the water mark (near the top of the stem) and the mark to which the instrument sinks in a 15% solution of salt. The space between these marks is divided into ten equal parts, and division of like sizes are continued up the stem.

These divisions are numbered upwards from the salt solution mark. A liquid is said to have a specific gravity of 17 degree Beaume "light", when the hydrometer sinks in it to mark number seventeen on this scale.

THE CALORIMETER

The apparatus employed in the measurment of the heat of vaporization is a calorimeter, containing a hollow spiral and inner receptacle. Vapor is sent into the spiral, where it is condensed, and the liquid thus produced gathers in the receiver, whence it is subsequently removed and weighed. The liquid heated in the vessel, whence

the vapor passes through the inner tube to the spiral. Here it condenses, warming the surrounding water of the calorimeter and is collected in the receptacle.

VISCOSIMETRY

The term viscosity represents the internal friction of an oil; it is the opposite of fluidity. It is usually determined by noting the time in seconds required for a definite quantity of oil to flow through cylindrical or conical openings, approximately 0.50 to 0.75 in. in length and 0.01 in. in diameter. The name or type of instrument used and the temperature should always be given in expressing viscosities numerically.

The viscosities of two oils can be roughly compared as follows: Two sample bottles containing the oils are held side by side and inverted. The oil that drops from its bottle first has a lower v'scosity than the other oil. Another crude method of comparing viscosities is to shake the sample bottles of the oils and then to note the rise through the oils of approximately equal size air bubble. The faster the movement of the bubble, the lower the viscosity of the oil.

A rough quantitive comparison can be obtained by filling a clean pipette with about 10 cu. cm. of the oil and counting the seconds required for the oil to flow from one mark on the upper stem of the pipette to another mark on the lower stem. Thereupon the pipette is cleaned with ether, dried, filled with the second oil and the experiment repeated. If the time of flow in seconds for one oil is one-half that of the other, the first oil has roughly one-half the viscosity, provided the temperatures of both oils during the test are the same. In order to be sure that both oils are at the same temperatures, they should be placed in beakers and left to stand side by side on a table for about one hour before the test is made.

For accurate results practical viscosimeters must be used. These instruments combine ruggedness of construction with rapidity of operation and work on the principle of permitting a small quantity of the liquid to flow through an orifice and noting the time of efflux.

Among the instruments in commercial use are the Engler Viscosimeter, which is specified for U. S. Navy fuel oil tests and is the standard instrument in Germany; the Sayboldt Universal Viscosimeter which is in general use in the United States; and the Redwood Viscosimeter, the standard instrument in England. These will now be briefly described.

Engler Viscosimeter: The Engler Viscosimeter consists of a cylindrical oil chamber provided with a concave bottom in the center of which is a conical orifice, 20 mm. (0.78 in.) in length and 2.9 mm. (0.114 in.) in diameter at the top of the orifice. This diameter tapers to 2.8 mm. (0.110 in.) at the bottom of the orifice. For standard work, the orifice is made of platinum, but ordinarily brass is used. The oil chamber is covered with an asbestos lagged lid.

A plug valve made of hard wood is used as stopper for the orifice.

The oil chamber is made of brass and on the inside is provided with three studs in a horizontal plane. These studs serve both to indicate the proper oil level and to assist in leveling the instrument.

Surrounding the oil chamber is a water bath. Water is used for temperatures up to about 120° Fahr. (about 50° C.) Above this temperature a heavy mineral oil is used instead of the water. The bath is heated by means of a ring gas burner.

Immediately below the oil tube outlet is a measuring flask graduated for 100 and 200 cu. cm. (6.1 and 12.2 cu. in.).

The method of use is as follows: 200 cu. cm. of water at 68° Fahr. (20° Cels.) are permitted to flow through the orifice and the time in seconds is counted. In the standard instrument this is 50 to 53 seconds. Thereupon the oil chamber is rinsed with alcohol, then ether, and finally dried. 200 cu. cm. of oil are introduced and brought to the proper temperature. Before introducing the oil, it should be strained in order to remove foreign particles, dirt and water. The oil is then allowed to flow through the orifice, and the time of flow noted. Suppose this is 360 seconds at 122° Fahr. (50° C.) and that the time of flow for 200 cu. cm. of water at 68° F. (20° C.) is 53 seconds, then:

Engler viscosity at

$$122° \text{ Fahr. } (50° \text{ C.}) = \frac{360}{53} = 6.8$$

If the oil had been tested at 302° Fahr. (150° Cels.) and the time noted as 90 sec., the viscosity would be:

Engler viscosity at

$$302° \text{ Fahr. } (150° \text{ Cels.}) = \frac{90}{53} = 1.7$$

In other words, with this instrument, a specific vicosity, which of course is a purely arbitrary number, is obtained by dividing the time in seconds required for 200 cu. cm. of oil at any temperature to flow through the orifice by the time in seconds required for 200 cu. cm. of water at 68° Fahr. (20° Cels.) to flow through the same orifice.

50 cu. cm. or 100 cu. cm. are frequently employed instead of 200 cu. cm. However, the times of flow must then be multiplied by 5 and 2.35 respectively in order to obtain results concordant with those determined in the ordinary way.

Redwood Viscosimeter: The Redwood Viscosimeter consists of a cylindrical oil chamber of 1 7/8 in. internal diameter and 3 1/2 in. depth. The oil chamber is made of copper, silvered on the inside and has a slightly concave bottom in the center of which is an agate jet or orifice 13 mm. (0.47 in.) in length and 1.7 mm. (0.067 in.) in diameter.

Surrounding the oil cylinder is the bath container made of copper and usually filled with water for temperature up to 200° Fahr. For higher

temperatures, a heavy mineral oil is used. The bath is provided with a copper heating tube set at a 45° angle and heated by a suitable gas burner. Uniform temperature is maintained in the bath by means of agitators or stirrers consisting of four light metal vanes fastened to a thin copper tube revolving around the oil chamber. The upper part of this copper tube has a broad curved flange which prevents the bath liquid from splashing into the oil chamber. The agitators are revolved by means of a handle.

The temperatures of both the oil and bath are determined by means of suitable thermometers suspended. The oil chamber stopper is a brass sphere which fits nicely into a hemispherical cavity in the agate jet. The brass ball is suspended by a wire.

A bracket gage or pointed stud determines the initial head of oil in the oil chamber and can be adjusted slightly in order to correct for variations in time of flow between different instruments due to unavoidable differences in the dimensions of the agate jets of Redwood Viscosimeters. A determination is made by bringing the oil and bath to the proper temperature, raising the ball valve and noting the time in seconds for 50 cu. cm. of the oil to pass through the jet. The viscosity is found as follows:

Redwood viscosity at

$$X^\circ \text{ F.} = \frac{T \times D \times 100}{535 \times 0.915}$$

in which

T = the time in seconds required for 50 cu. cm. of oil at X° F. to flow through the orifice.

D = the specific gravity of the oil at X° F.

535 = the time in seconds required for 50 cu. cm. of refined rape-seed oil at 60° Fahr. to flow through the orifice.

0.915 = the specific gravity of rape-seed oil at 60° Fahr. referred to water at 60° Fahr. as unity.

For water, the rate of flow of 50 cu. cm. at 60° F. is about 25.5 seconds. Before being used in the instrument, the oil should be strained through a fine wire gauze or piece of muslin.

Sayboldt Viscosimeter: With the exception of two glass windows in the bath container and a glass tube in the pipette, the Sayboldt Viscosimeter is made entirely of metal. The oil chamber or cylinder has about 83 cu. cm. (5.05 cu. in.) capacity and the upper portion of the cylinder is perforated by a ring of small holes through which excess oil flows into a fixed gallery.

The jet in the bottom of the cylindrical oil chamber consists of an outside metal tube within which is a glass tube. At the bottom of this tube is the orifice, 9 mm. (0.354 in.) in length and 1.6 mm. (0.063 in.) in internal diameter. On the outside of the metal tube is a flange which

is secured by screws to a flange on a short tube soldered to the bottom of the bath container.

Thermomters are provided for measuring the temperatures of the oil and the bath. The viscosity is determined by noting the time in seconds required for 60 cu. cm. of liquid to flow through the orifice. For water at 70° Fahr. this is 30 sec. in a standard instrument.

Sayboldt viscosity tests are generally made at the following temperatures:

210° Fahrenheit for valve oils.

100° Fahrenheit for machine oils.

100° Fahrenheit for motor oils used in internal combustion engines.

Sayboldt viscosities may be converted into Engler and Redwood viscosities, and also into readings of other instruments by means of the factors given in following table:

FACTORS TO REDUCE SAYBOLDT TIMES TO READINGS IN OTHER INSTRUMENTS

Viscosimeter	70° F.	100° F.	212° F.	338° F.
McMichael	0.50	0.55	0.60	0.65
Sayboldt "A"	0.50	1.00	---	---
Sayboldt "C"	---	---	0.46	0.72
Engler	0.035	0.030	0.028	0.027
Tagliabue	0.25	0.28	0.51	---
Penn. R. R. Pipett	0.30	0.47	0.51	0.94
Scott	0.13	0.13	---	---
Redwood	0.83	0.85	0.88	0.90
Magruder Plunger	1.25	1.04	2.00	---
Ostwald	1.90	1.85	1.68	1.30

To proceed now, if the Sayboldt viscosity of an oil at 100° F. is 100 sec., the Redwood viscosity is found by multiplying this by 0.85, thereby giving 85 as the Redwood viscosity at 100° F. similarly, the Engler number is found by multiplying the Sayboldt viscosity by 0.03, giving 3.0 as the Engler number at 100° F. These conversion factors are not exact, as they vary greatly with the actual viscosities, and are merely given here to demonstrate the existing differences in actual results.

CRITICAL TEMPERATURES

When a liquid and its vapor confined in a vessel are heated, a portion of the liquid vaporizes, the pressure increases, the density of the vapor increases, and the density of the liquid decreases. When a certain temperature is reached, the density of the liquid and of the vapor become identical, and the vapor and the liquid are physically identical. This temperature is called "critical temperature" of the liquid.

The heat of vaporization of a liquid is less, the higher the temperature (and pressure) at which the vaporization takes place, and becomes zero at the critical temperature. For example:

The heat of vaporization of water is 606.5 at 0 degree, 539.9 at 100 degree, and 464.3 at 200 degree.

In the following tables the critical temperatures of certain substances are given:

CRITICAL TEMPERATURES OF VARIOUS LIQUIDS

	Deg. C.		Deg. C.
Alcohol (ethyl)	240	Methane (CH_4)	81
Ammonia (NH_3)	130	Nitrogen	146
Benzol	280	Nitrous oxide (N_2O)	35.4
Bromine	302	Oxygen	118
Carbon monoxide (CO)	141	Sulphuretted hydrogen (H_2S)	100
Carbon dioxide (CO_2)	31	Sulphur dioxide (SO_2)	156
Chlorine	141	Turpentine oil	376
Chloroform	260	Water	365

HEATING FUEL OIL

Fuel oil is generally preheated in order to reduce the viscosity and therefore to assist the atomization of the oil in oil injection devices. There is no practical gain in heating the oil above the temperature corresponding to a viscosity for which the atomizers produce efficient atomization. This viscosity usually is 8 Engler. In the case of the more viscous oils, care should be taken that the oil is not heated above the flashpoint. Unless leaks in the oil lines are guarded against, excessive preheating to reduce the viscosity would be dangerous expedient. A temperature of 300° F. is frequently sufficient for fine atomization of the heaviest oils and for their complete and perfect combustion in the engine.

For proper atomization, the oil should be preheated to the temperature given in following table:

PREHEATING TEMPERATURE FOR FUEL OIL

Degree Baume	Specific Gravity 60°/60° F.*	Temp. Degr. F.
12.0	0.9859	300
16.0	0.9589	250
20.0	0.9333	200

Note: * Indicates that the specific gravity of the oil at 60° Fahrenheit is referred to water at 60° Fahrenheit as unity.

DENSITY AND VISCOSITY OF WATER AT DIFFERENT TEMPERATURES

Experiments by Stanton on the flow of air through pipes of 5 and 7.4 centimeters diameter show that for exact similarity of distribution of velocity over the cross-section of the pipe the center velocities should be inversely proportional to the pipe diameters, that is, that the ratio of average velocity to center velocity will be the same in two pipes of different diameters if this condition be satisfied. A comparison by E. Buckingham of the experiments by Stanton and by Saph and Schroder on flow through brass tubes indicates that to secure similar flow conditions with fluids of different densities and viscosities the velocities and diameters should be so selected as to satisfy the equation:

$$\frac{V\,a\,d}{m} = \text{a constant}$$

in which V is the central velocity, a radius of the pipe, d the density of the fluid, and m the coefficient of viscosity. The density and viscosity of water at different temperatures are given in the following table:

Temp.	Wt. per Cu. Ft. Lb.	Coefficient Viscosity Dynes per Sq. Cm.	Temp.	Wt. per Cu. Ft. Lb.	Coefficient Viscosity Dynes per Sq. Cm.
32	62.42	.0179	130	61.56	.0051
40	62.42	.0155	140	61.37	.0047
50	62.41	.0131	150	61.18	.0043
60	62.37	.0112	160	60.98	.0040
70	62.31	.0097	170	60.77	.0037
80	62.23	.0086	180	60.55	.0035
90	62.13	.0077	190	60.32	.0032
100	62.02	.0068	200	60.07	.0030
110	61.89	.0062	210	59.82	.0028
120	61.74	.0056			

UNIT OF HEAT

The quantity measure of heat is the thermal unit. The British thermal unit (as distinguished from the French thermal unit, or calorie) is that quantity of heat which is required to raise the temperature of one pound of pure water through one degree Fahrenheit, at or near 39.1° Fahrenheit, the temperature of maximum density of water. As employed in general practice, the term is usually abbreviated to "B. T. U."

The relation existing between the temperature of water in degrees Fahrenheit and the number of thermal units contained therein, together with the increase in the number of thermal units for each increment of temperature of 5 degrees, is indicated in following table.

NUMBER OF THERMAL UNITS CONTAINED IN ONE POUND OF WATER

Temp. Degrees F.	No of Thermal Units	Increase	Temp. Degrees F.	No of Thermal Units	Increase
35	35.000	----	215	215.939	5.065
40	40.001	5.001	220	221.007	5.068
45	45.002	5.001	225	226.078	5.071
50	50.003	5.001	230	231.153	5.075
55	55.006	5.003	235	236.232	5.079
60	60.009	5.003	240	241.313	5.081
65	65.014	5.005	245	246.398	5.085
70	70.020	5.006	250	251.487	5.089
75	75.027	5.007	255	256.579	5.092
80	80.036	5.009	260	261.674	5.095
85	85.045	5.009	265	266.774	5.100
90	90.055	5.010	270	271.878	5.104
95	95.067	5.012	275	276.985	5.107
100	100.080	5.013	280	282.095	5.110
105	105.095	5.015	285	287.210	5.115
110	110.110	5.015	290	292.329	5.119
115	115.129	5.019	295	297.452	5.123
120	120.149	5.020	300	302.580	5.128
125	125.169	5.020	305	307.712	5.132
130	130.192	5.023	310	312.848	5.136
135	135.217	5.025	315	317.988	5.140
140	140.245	5.028	320	323.134	5.146
145	145.175	5.030	325	328.284	5.150
150	150.305	5.030	330	333.438	5.154
155	155.339	5.034	335	338.596	5.158
160	160.374	5.035	340	343.750	5.163
165	165.413	5.039	345	348.927	5.168
170	170.453	5.040	350	354.101	5.174
175	175.497	5.044	355	359.280	5.179
180	180.542	5.045	360	364.464	5.184
185	185.591	5.049	365	369.653	5.189
190	190.643	5.052	370	374.846	5.193
195	195.697	5.054	375	380.044	5.198
200	200.753	5.056	380	385.247	5.203
205	205.813	5.060	385	390.456	5.209
210	210.874	5.061	390	395.672	5.216

ANALYSIS OF SEA WATER

Salt water is known to be a solvent of iron or steel, and when brought to a stage of high heating temperature in the exhaust of Internal Combustion Engines, the magnesium chloride, about 250 grains of which are contained in every gallon, becomes highly corrosive.

TABLE OF SOLVING SUBSTANCES

Carbonate of lime	9.79 grains per gal.
Sulphate of lime	114.36 grains per gal.
Sulphate of magnesium	134.86 grains per gal.
Chloride of magnesium	244.46 grains per gal.
Chloride of sodium	1706.00 grains per gal.
Total solids	2209.47 grains per gal.

OFFICIAL TEMPERATURES AND CRITICAL PRESSURES

Substances	Critical Temperature °C	Critical Temperature °F	Critical Pressures in Atmospheres
Alcohol	235	455	64
Ammonia (NH_3)	130	266	115
Corbon dioxide (CO_2)	31	88	73
Carbon disulphide (CS_2)	273	523	73
Ether	195	383	36
Hydrogen	—235	—391	20
Nitrogen	—146	—231	33
Oxygen	—118	—180	50
Sulphur dioxide (SO_2)	155	311	79
Water	365	689	200

HYDROMETER SCALES

Barkometer Degrees = the first three figures of the decimal of the corresponding specific gravity, thus:

1.008 specific gravity = 8 degrees Barkometer.
1.015 " " = 15 " "
1.223 " " = 223 " "

Twaddell Degrees = the first three figures of the decimal of the corresponding specific gravity, divided by five, thus:

1.010 specific gravity = 2 degrees Twaddell.
1.125 " " = 25 " "

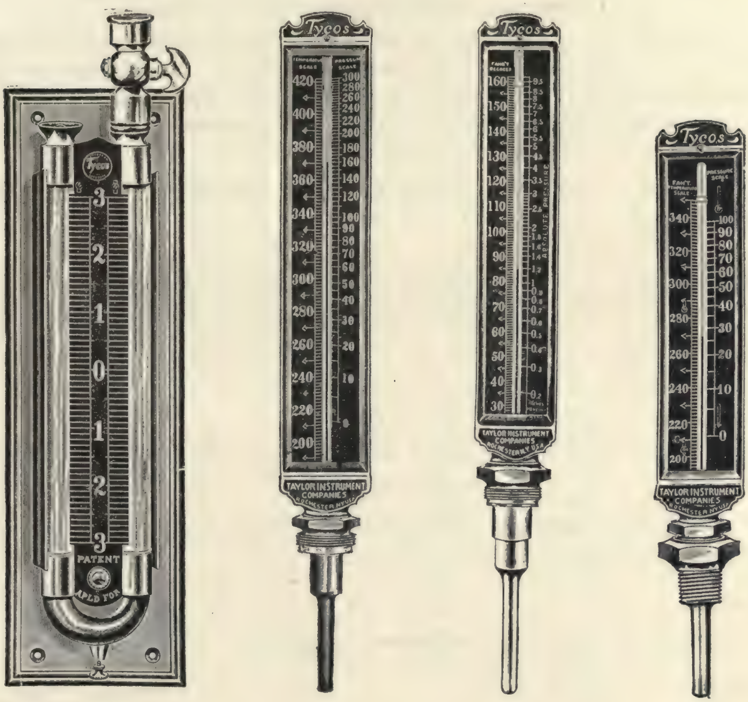
Densimetric Degrees = the first two figures of the decimal of the corresponding specific gravity, thus:

1.018 specific gravity = 1.8 degrees Densimetric.
1.234 " " = 23.4 " "

Brix Degrees = sugar percentages = 1.8° Baume (about).

SPECIFIC HEAT OF WATER

Temperature Degrees F.	Specific Heat at Given Temp. Freezing Point = 1 degree.	Temperature Degrees F.	Specific Heat at Given Temp. Freezing Point = 1 degree.
32	1.0000	248	1.0177
50	1.0005	266	1.0204
68	1.0012	284	1.0252
86	1.0020	302	1.0262
104	1.0030	320	1.0294
122	1.0042	338	1.0328
140	1.0056	356	1.0364
158	1.0072	374	1.0401
176	1.0089	394	1.0440
194	1.0109	410	1.0481
212	1.0130	428	1.0524
230	1.0153	446	1.0568



(*a*) (*b*) (*c*) (*d*)

"Tyco" Instruments—(a) Draft Gauge. (b) High Pressure Thermometer. (c) Vacuum Gauge. (d) Low Pressure Thermo Gauge.

HOW A THERMOMETER IS GRADUATED

A mercurial tube is placed in melting ice, which is the temperature at which water freezes or ice melts. When the mercury has fallen to its lowest point it is marked on the tube. Then it is placed in boiling water under atmospheric pressure at sea level. When the mercurial column reaches a certain height it remains there until taken out, this point is marked. Now we have the freezing and boiling points and we can graduate the thermometer either Centigrade (English Thermometer) or Fahrenheit.

If we graduate it Fahrenheit, we will let the freeging point equal 32° and the boiling point 212°, then we have 212 — 32 = 180° between freezing and boiling; then the space between 212 degrees and 32 degrees will be equally divided into 180 parts, each part representing a degree. On the Centigrade, we will let 0 equal the freezing point and 100° the boiling point, then there is 100 degrees between the freezing and boiling point on the Centigrade; then 1 degree of Fahrenheit is equal to 100 ÷ 180, or 5/9° Centigrade.

Fahrenheit and Centigrade Thermometers

Everyone is familiar with the Fahrenheit Thermometer and its readings, but comparatively few are familiar with the significance of the Centigrade Scale. At the present time many of the heating instruments are graduated to the Centigrade readings. To convert the readings from one scale to the other, use the following rule:

Subtract 32 from the Fahrenheit, divide the remainder by 9 and multiply by 5.

Example:

—Convert 212° Fahrenheit to Centigrade equivalent.

Solution:

$212 - 32 = 180$

$180 \div 9 = 20$

$20 \times 5 = 100°$ C., or the respective readings of the temparature at which water boils.

To convert Centigrade to Fahrenheit readings, divide the Centigrade readings by 5, multiply by 9 and add 32.

Example:

Convert 100° C. to Fahrenheit equivalent.

Solution:

$100 \div 5 = 20$

$20 \times 9 = 180$

$180 + 32 = 212°$ F.

SPECIFIC GRAVITIES IN DEGREES BEAUME: LIQUIDS LIGHTER THAN WATER

Degree Baume	Specific Gravity	Degree Baume	Specific Gravity	Degree Baume	Specific Gravity
10	1.000	25	0.903	40	0.824
11	0.993	26	0.897	41	0.819
12	0.986	27	0.892	42	0.814
13	0.979	28	0.886	43	0.805
14	0.972	29	0.881	46	0.796
15	0.966	30	0.875	48	0.787
16	0.959	31	0.870	50	0.778
17	0.952	32	0.864	52	0.769
18	0.946	33	0.859	54	0.761
19	0.940	34	0.854	56	0.753
20	0.933	35	0.849	58	0.745
21	0.927	36	0.843	60	0.737
22	0.921	37	0.838	65	0.718
23	0.915	38	0.833	70	0.700
24	0.909	39	0.828	75	0.683

THE BEAUME SCALE:

The usual method of indicating the weight of crude oil for fuel is by the Beaume Scale. The numbers of this scale are given by the formula:

$$\text{Degree Beaume} = \frac{140}{\text{S.G.}} - 130;$$ where S. G. is the specific gravity, water being — 1.

In the following table the specific gravity and weight per gallon for the different degrees on the Beaume scale are given:

Degrees Beaume	Pounds per U.S. Gallon	Specific Gravity	Degrees Beaume	Pounds per U.S. Gallon	Specific Gravity
10	8.336	1.000	26	7.477	.897
11	8.277	.993	27	7.435	.892
12	8.219	.986	28	7.385	.886
13	8.161	.979	29	7.344	.881
14	8.102	.972	30	7.294	.875
15	8.052	.966	31	7.252	.870
16	7.994	.959	32	7.202	.864
17	7.935	.952	33	7.160	.859
18	7.885	.946	34	7.119	.854
19	7.835	.940	35	7.069	.848
20	7.777	.933	36	7.027	.843
21	7.727	.927	37	6.985	.838
22	7.677	.921	38	6.944	.833
23	7.627	.915	39	6.902	.828
24	7.577	.909	40	6.869	.824
25	7.527	.903			

SPECIFIC GRAVITIES IN DEGREES BAUME AND TWADDLE.
LIQUIDS HEAVIER THAN WATER.

Hydrometer Reading —Degrees— Twaddle	Baume	Specific Gravity	Hydrometer Reading —Degrees— Twaddle	Baume	Specific Gravity
0	.0	1.000	40	24.0	1.200
1	.7	1.005	41	24.5	1.205
2	1.4	1.010	42	25.0	1.210
3	2.1	1.015	43	25.5	1.215
4	2.7	1.020	44	26.0	1.220
5	3.4	1.025	45	26.4	1.225
6	4.1	1.030	46	26.9	1.230
7	4.7	1.035	47	27.4	1.235
8	5.4	1.040	48	27.9	1.240
9	6.0	1.045	49	28.4	1.245
10	6.7	1.050	50	28.8	1.250
11	7.4	1.055	51	29.3	1.255
12	8.0	1.060	52	29.7	1.260
13	8.7	1.065	53	30.2	1.265
14	9.4	1.070	54	30.6	1.270
15	10.0	1.075	55	31.1	1.275
16	10.6	1.080	56	31.5	1.280
17	11.2	1.085	57	32.0	1.285
18	11.9	1.090	58	32.4	1.290
19	12.4	1.095	59	32.8	1.295
20	13.0	1.100	60	33.3	1.300
21	13.6	1.105	61	33.7	1.305
22	14.2	1.110	62	34.2	1.310
23	14.9	1.115	63	34.6	1.315
24	15.4	1.120	64	35.0	1.320
25	16.0	1.125	65	35.4	1.325
26	16.5	1.130	66	35.8	1.330
27	17.1	1.135	67	36.2	1.335
28	17.7	1.140	68	36.6	1.340
29	18.3	1.145	69	37.0	1.345
30	18.8	1.150	70	37.4	1.350
31	19.3	1.155	71	37.8	1.355
32	19.8	1.160	72	38.2	1.360
33	20.3	1.165	73	38.6	1.365
34	20.9	1.170	74	39.0	1.370
35	21.4	1.175	75	39.4	1.375
36	22.0	1.180	76	39.8	1.380
37	22.5	1.185	77	40.1	1.385
38	23.0	1.190	78	40.5	1.390
39	23.5	1.195	79	40.8	1.395

SPECIFIC GRAVITIES IN DEGREES BAUME AND TWADDLE. LIQUIDS HEAVIER THAN WATER.

Hydrometer Reading —Degrees— Twaddle	Baume	Specific Gravity	Hydrometer Reading —Degrees— Twaddle	Baume	Specific Gravity
80	41.2	1.400	120	54.1	1.600
81	41.6	1.405	121	54.4	1.605
82	42.0	1.410	122	54.7	1.610
83	42.3	1.415	123	55.0	1.615
84	42.7	1.420	124	55.2	1.620
85	43.1	1.425	125	55.5	1.625
86	43.4	1.430	126	55.8	1.630
87	43.8	1.435	127	56.0	1.635
88	44.1	1.440	128	56.3	1.640
89	44.4	1.445	129	56.6	1.645
90	44.8	1.450	130	56.9	1.650
91	45.1	1.455	131	57.1	1.655
92	45.4	1.460	132	57.4	1.660
93	45.8	1.465	133	57.7	1.665
94	46.1	1.470	134	57.9	1.670
95	46.4	1.475	135	58.2	1.675
96	46.7	1.480	136	58.4	1.680
97	47.1	1.485	137	58.7	1.685
98	47.4	1.490	138	58.9	1.690
99	47.8	1.495	139	59.2	1.695
100	48.1	1.500	140	59.5	1.700
101	48.4	1.505	141	59.7	1.705
102	48.7	1.510	142	60.0	1.710
103	49.0	1.515	143	60.2	1.715
104	49.4	1.520	144	60.4	1.720
105	49.7	1.525	145	60.6	1.725
106	50.0	1.530	146	60.9	1.730
107	50.3	1.535	147	61.1	1.735
108	50.6	1.540	148	61.4	1.740
109	50.9	1.545	149	61.6	1.745
110	51.2	1.550	150	61.8	1.750
111	51.5	1.555	151	62.1	1.755
112	51.8	1.560	152	62.3	1.760
113	52.1	1.565	153	62.5	1.765
114	52.4	1.570	154	62.8	1.770
115	52.7	1.575	155	63.0	1.775
116	53.0	1.580	156	63.2	1.780
117	53.3	1.585	157	63.5	1.785
118	53 6	1.590	158	63.7	1.790
119	53.9	1.595	159	64.0	1.795

SPECIFIC GRAVITIES IN DEGREES BAUME AND TWADDLE. LIQUIDS HEAVIER THAN WATER.

Hydrometer Reading —Degrees— Twaddle	Baume	Specific Gravity	Hydrometer Reading —Degrees— Twaddle	Baume	Specific Gravity
160	64.2	1.800	165	65.2	1.825
161	64.4	1.805	166	65.5	1.830
162	64.6	1.810	167	65.7	1.835
163	64.8	1.815	168	65.9	1.840
164	65.0	1.820	169	66.1	1.845
			170	66.3	1.850
			171	66.5	1.855

TABLE OF EQUIVALENT FOR FUEL OIL

This chart shows the equivalents for fuel oils at various gravities and is taken at 60° Fahrenheit. Naturally, a temperature adjustment must be made to determine true specific gravity. This adjustment is as follows:

For every degree above 60° F., subtract .0004
For every degree below 60° F., add .0004

Specific Gravity	Beaume Gravity	Lbs. per Amer. Gal.	Lbs per English Gal.	Cu. Ft. Amer. per Ton	Gal. Amer. per Ton	Gal. English per Ton	Bbls. Amer. per Ton
1.0000	10.	8.331	10.	35.94	268.875	224.	6.40
.9956	10.5	8.302	9.995	36.09	269.81	224.75	6.42
.9930	11.	8.273	9.930	36.19	270.76	225.55	6.44
.9895	11.5	8.244	9.895	36.32	271.71	226.33	6.46
.9860	12.	8.214	9.860	36.45	272.67	227.13	6.49
.9825	12.5	8.185	9.825	35.57	273.66	227.96	6.51
.9790	13.	8.156	9.790	36.71	274.62	228.80	6.54
.9755	13.5	8.127	9.705	36.84	275.62	229.62	6.56
.9720	14.	8.098	9.702	36.97	276.67	230.49	6.58
.9685	14.5	8.069	9.685	37.10	277.47	231.16	6.60
.9655	15.	8.044	9.650	37.22	278.46	231.98	6.63
.9625	15.5	8.019	9.625	37.34	279.33	232.71	6.65
.9695	16.	7.994	9.595	37.46	280.19	233.42	6.66
.9560	16.5	7.964	9.560	37.59	281.26	234.31	6.69
.9530	17.	7.929	9.530	37.71	282.22	235.11	6.74
.9495	17.5	7.910	9.495	37.85	283.08	235.90	6.75
.9465	18.	7.885	9.465	37.97	284.08	236.66	6.76
.9430	18.5	7.856	9.430	38.11	285.13	257.52	6.76
.9400	19.	7.831	9.400	38.23	286.04	238.30	6.81
.9370	19.5	7.806	9.370	38.35	286.95	239.06	6.83
.9340	20.	7.781	9.340	38.47	287.88	239.82	6.85
.9310	20.5	7.756	9.310	38.60	288.88	240.60	6.87

TABLE EQUIVALENT FOR FUEL OIL

Specific Gravity	Beaume Gravity	Lbs. per Amer. Gal.	Lbs per English Gal.	Cu. Ft. Amer. per Ton	Gal. Amer. per Ton	Gal. English per Ton	Bbls. Amer. per Ton
.9280	21.	7.730	9.280	38.73	289.74	241.34	6.89
.9250	21.5	7.706	9.250	38.85	290.68	242.16	6.89
.9220	22.	7.680	9.220	38.98	291.62	242.95	6.94
.9195	22.5	7.660	9.195	39.09	292.42	243.61	6.96
.9165	23.	7.635	9.165	39.21	293.25	244.40	6.98
.9135	23.5	7.615	9.135	39.34	294.15	245.21	7.00
.9105	24.	7.585	9.105	39.47	295.31	246.01	7.03
.9045	25.	7.536	9.040	39.73	297.24	247.64	7.07
.8990	26.	7.490	8.990	39.97	299.06	249.15	7.08
.8930	27.	7.440	8.930	40.24	301.07	250.84	7.12
.8870	28.	7.390	8.870	40.51	303.11	252.53	7.21
.8815	29.	7.344	8.815	40.77	305.01	254.00	7.26
.8755	30.	7.294	8.755	41.04	307.10	255.85	7.31
.8700	31.	7.248	8.700	41.31	309.19	257.47	7.36
.8650	32.	7.206	8.650	41.54	310.85	258.94	7.40
.8595	33.	7.160	8.595	41.81	312.84	260.61	7.44
.8545	34.	7.119	8.545	42.05	314.65	262.14	7.46
.8490	35.	7.070	8.490	42.32	316.83	263.83	7.54
.8440	36.	7.031	8.440	42.58	318.58	265.40	7.58
.8395	37.	6.994	8.395	42.81	320.27	266.82	7.62
.8345	38.	6.952	8.345	43.06	322.67	268.42	7.70
.8295	39.	6.911	8.295	43.32	324.12	270.04	7.71
.8250	40.	6.873	8.250	43.56	325.90	271.51	7.78

RELATIVE COST OF COAL AND OIL

The primary object in giving this table is to draw an approximate comparison in cost of coal as used in generation of steam in contrast to oil used in Diesel Engines for fuels. It is understood, that the variation of either coal, as well as oil, in prices, average about from 12 to 14%. The following tables will give the average prevailing fuel cost of either coal or oil.

Oil cents per gallon	Oil dollars per barrel	Coal dollars per ton	Oil cents per gallon	Oil dollars per barrel	Coal dollars per ton
2.00	$0.82	$3.92	3.25	$1.33	$6.37
2.25	0.92	4.41	3.50	1.43	6.86
2.50	1.02	4.90	4.00	1.64	7.84
2.75	1.13	5.39	4.50	1.84	8.82
3.00	1.23	5.88	5.00	2.05	9.80

CONVERSION TABLE FOR DEGREES BAUME (LIGHTER THAN WATER) TO SPECIFIC GRAVITY AND LBS. PER GALLON

Degrees Baume	Specific Gravity	Pounds in 1 Gallon (American)	Degrees Baume	Specific Gravity	Pounds in 1 Gallon (American)
10	1.0000	8.33	43	.8092	6.74
11	.9929	8.27	44	.8045	6.70
12	.9859	8.21	45	.8000	6.66
13	.9790	8.16	46	.7954	6.63
14	.9722	8.10	47	.7909	6.59
15	.9655	8.04	48	.7865	6.55
16	.9589	7.99	49	.7831	6.52
17	.9523	7.93	50	.7777	6.48
18	.9459	7.88	51	.7734	6.44
19	.9395	7.83	52	.7692	6.41
20	.9333	7.78	53	.7650	6.37
21	.9271	7.72	54	.7608	6.34
22	.9210	7.67	55	.7567	6.30
23	.9150	7.62	56	.7526	6.27
24	.9090	7.57	57	.7486	6.24
25	.9032	7.53	58	.7446	6.20
26	.8974	7.48	59	.7407	6.17
27	.8917	7.43	60	.7368	6.14
28	.8860	7.38	61	.7329	6.11
29	.8805	7.34	62	.7290	6.07
30	.8750	7.29	63	.7253	6.04
31	.8695	7.24	64	.7216	6.01
32	.8641	7.20	65	.7179	5.98
33	.8588	7.15	66	.7142	5.95
34	.8536	7.11	67	.7106	5.92
35	.8484	7.07	68	.7070	5.89
36	.8433	7.03	69	.7035	5.86
37	.8383	6.98	70	.7000	5.83
38	.8333	6.94	75	.6829	5.69
39	.8284	6.90	80	.6666	5.55
40	.8235	6.86	85	.6511	5.42
41	.8187	6.82	90	.6363	5.30
42	.8139	6.78	95	.6222	5.18

TABLE OF GALLONS

	Cubic In. in a Gallon	Weight of a Gal. in lbs. Avoirdupois	Gallons in a Cubic Foot
United States	231	8.33	7.480
New York	231.819	8.00	7.901
Imperial	277.274	10.00	6.232

CONVENIENT TABLE TO ESTABLISH POUNDS PER SQUARE INCHES TO HEAD IN FEET

For Liquids at 62° Fahrenheit, Weighing 62.364 lb. Per Cu. Ft.

Pounds	Head	Pounds	Head	84	194.0
per	in	per	in	85	196.3
Sq. In.	Feet	Sq. In.	Feet	86	198.6
2	4.619	43	99.31	87	200.9
3	6.928	44	101.6	88	203.2
4	9.238	45	103.9	89	205.5
5	11.55	46	106.2	90	207.8
6	13.86	47	108.5	91	210.2
7	16.17	48	110.8	92	212.5
8	18.48	49	113.2	93	214.8
9	20.78	50	115.5	94	217.1
10	23.09	51	117.8	95	219.4
11	25.40	52	120.1	96	221.7
12	27.71	53	122.4	97	224.0
13	30.02	54	124.7	98	226.3
14	32.33	55	127.0	99	228.6
15	34.64	56	129.3	100	230.9
16	36.95	57	131.6	105	242.4
17	39.26	58	133.9	110	254.0
18	41.57	59	136.3	115	265.5
19	43.88	60	138.6	120	277.1
20	46.19	61	140.9	125	288.6
21	48.50	62	143.2	130	300.2
22	50.81	63	145.5	135	311.7
23	53.12	64	147.8	140	323.3
24	55.43	65	150.1	145	334.8
25	57.74	66	152.4	150	346.4
26	60.05	67	154.7	155	357.9
27	62.36	68	157.0	160	369.5
28	64.66	69	159.3	165	381.0
29	66.97	70	161.7	170	392.6
30	69.28	71	163.0	175	404.1
31	71.59	72	166.3	190	438.8
32	73.90	73	168.6	195	450.3
33	76.21	74	170.9	200	461.9
34	78.52	75	173.2	210	485.0
35	80.83	76	175.5	220	508.1
36	83.14	77	177.8	230	531.2
37	85.45	78	180.1	240	554.3
38	87.76	79	182.4	250	577.4
39	90.07	80	184.8	260	600.5
40	92.38	81	187.1	270	623.6
41	94.69	82	189.4	280	646.6
42	97.00	83	191.7	290	669.7

Pounds per Sq. In.	Head in Feet	Pounds per Sq. In.	Head in Feet	Pounds per Sq. In.	Head in Feet
300	692.8	370	854.5	440	1016.
310	715.9	380	877.6	450	1039.
320	739.0	390	900.7	460	1062.
330	762.1	400	923.8	470	1085.
340	785.2	410	946.9	480	1108.
350	808.3	420	970.0	490	1132.
360	831.4	430	993.1	500	1155.

UNITED STATES GALLONS IN ROUND TANKS FOR ONE FOOT IN DEPTH

Diam of Tanks Ft.	In.	No. U. S. Gals.	Cub. Ft. and area in sq. ft.	Diam of Tanks Ft.	In.	No. U. S. Gals.	Cub. Ft. and area in sq. ft.
1	0	5.87	.785	3	5	68.58	9.168
1	1	6.89	.922	3	6	71.97	9.621
1	2	8.00	1.069	3	7	75.44	10.085
1	3	9.18	1.227	3	8	78.99	10.559
1	4	10.44	1.396	3	9	82.62	11.045
1	5	11.79	1.576	3	10	86.33	11.541
1	6	13.22	1.767	3	11	90.13	12.048
1	7	14.73	1.969	4	0	94.00	12.566
1	8	16.32	2.182	4	1	97.96	13.095
1	9	17.99	2.405	4	2	102.00	13.635
1	10	19.75	2.640	4	3	106.12	14.186
1	11	21.58	2.885	4	4	110.32	14.748
2	0	23.50	3.142	4	5	114.61	15.321
2	1	25.50	3.409	4	6	118.97	15.90
2	2	27.58	3.687	4	7	123.42	16.50
2	3	29.74	3.976	4	8	127.95	17.10
2	4	31.99	4.276	4	9	132.56	17.72
2	5	34.31	4.587	4	10	137.25	18.35
2	6	36.72	4.909	4	11	142.02	18.99
2	7	39.21	5.241	5	0	146.88	19.63
2	8	41.78	5.585	5	1	151.82	20.29
2	9	44.38	5.940	5	2	156.83	20.97
2	10	47.16	6.305	5	3	161.93	21.65
2	11	49.98	6.581	5	4	167.12	22.34
3	0	52.88	7.069	5	5	172.38	23.04
3	1	55.86	7.467	6	6	177.72	23.76
3	2	58.92	7.876	5	7	183.15	24.48
3	3	62.06	8.296	5	8	188.66	25.22
3	4	65.28	8.727	5	9	194.25	25.97

Diam of Tanks Ft.	In.	No. U. S. Gals.	Cub. Ft. and area in sq. ft.	Diam of Tanks Ft.	In.	No. U. S. Gals.	Cub. Ft. and area in sq. ft.
5	10	199.92	26.73	16	9	1648.40	220.35
5	11	205.67	27.49	17	0	1697.90	226.98
6	0	211.51	28.27	17	3	1748.20	233.71
6	3	229.50	30.68	17	6	1799.30	240.35
6	6	248.23	33.18	17	9	1851.10	247.45
6	9	267.69	35.78	18	0	1903.60	254.47
7	0	287.88	38.48	18	3	1956.80	261.59
7	3	308.81	41.28	18	6	2010.80	268.80
7	6	330.48	44.18	18	9	2065.50	276.12
7	9	352.88	47.17	19	0	2120.90	283.53
8	0	376.01	50.27	19	3	2177.10	291.04
8	3	399.88	53.46	19	6	2234.00	298.65
8	6	424.48	56.75	19	9	2291.70	306.35
8	9	449.82	60.13	20	0	2350.10	314.16
9	0	475.89	63.62	20	3	2409.20	322.06
9	3	502.70	67.20	20	6	2469.10	330.06
9	6	530.24	70.88	20	9	2529.60	338.16
9	9	558.51	74.66	21	0	2591.00	346.36
10	0	587.52	78.54	21	3	2653.00	354.66
10	3	617.26	82.52	21	6	2715.80	363.05
10	6	640.74	86.59	21	9	2779.30	371.54
10	9	678.95	90.76	22	0	2843.60	380.13
11	0	710.90	95.03	22	3	2908.60	388.82
11	3	743.58	99.40	22	6	2974.30	397.61
11	6	776.99	103.87	22	9	3040.80	406.49
11	9	811.14	108.43	23	0	3108.00	415.48
12	0	846.03	113.10	23	3	3175.90	424.56
12	3	881.65	117.86	23	6	3244.60	433.74
12	6	918.00	122.72	23	9	3314.00	443.01
12	9	955.09	127.68	24	0	3384.10	452.39
13	0	992.91	132.73	24	3	3455.00	461.86
13	3	1031.50	137.89	24	6	3526.60	471.44
13	6	1070.80	143.14	24	9	3598.90	481.11
13	9	1110.80	148.49	25	0	3672.00	490.87
14	0	1151.50	153.94	25	3	3745.80	500.74
14	3	1193.00	159.48	25	6	3820.30	510.71
14	6	1235.30	165.13	25	9	3895.60	520.77
14	9	1278.20	170.87	26	0	3971.60	530.93
15	0	1321.90	176.71	26	3	4048.40	541.19
15	3	1366.40	182.65	26	6	4125.90	551.55
15	6	1411.50	188.69	26	9	4204.10	562.00
15	9	1457.40	194.83	27	0	4283.00	572.66
16	0	1504.10	201.06	27	3	4362.70	583.21
16	3	1551.40	207.39	27	6	4443.10	593.96
16	6	1599.50	213.82	27	9	4524.30	604.81

Diam of Tanks Ft.	In.	No. U. S. Gals.	Cub. Ft. and area in sq. ft.	Diam of Tanks Ft.	In.	No. U. S. Gals.	Cub. Ft. and area in sq. ft.
28	0	4606.20	615.75	30	9	5555.40	742.64
28	3	4688.80	626.80	31	0	5646.10	754.77
28	6	4772.10	637.94	31	3	5737.50	766.99
28	9	4856.20	649.18	31	6	5829.70	779.31
29	0	4941.00	660.52	31	9	5922.60	791.73
29	3	5026.60	671.96	32	0	6016.20	804.25
29	6	5112.90	683.49	32	3	6110.60	816.86
29	9	5199.90	695.13	32	6	6205.70	829.58
30	0	5287.70	706.86	32	9	6301.50	842.39
30	3	5376.20	718.69				
30	6	5465.40	730.62				

31½ gallons equal 1 barrel.

NOTE: To find the capacity of tanks greater than the largest given in the table, look in the table for a tank of one-half of the given size and multiply its capacity by 4, or one of one-third its size and multiply its capacity by 9, etc.

CO_2 AND FUEL LOSSES.

CALCULATED ON FOLLOWING CONDITIONS:

Oil as used for fuel—18633 B. T. U., 84.73% carbon, 11.74% hydrogen, 1.06% sulphur, 5% nitrogen, .87% oxygen, .7% moisture and .4% sediment.

Atmospheric temperature 55% F., humidity 88, exhaust temperature 500° F., Kern Oil 16° B.

Per Cent CO_2	Per Cent Excess Air	B. T. U. Loss	Per Cent Preventable Fuel Loss
15.6	0	0	.0
15	5	75	.4
14	10	186	1.
13	18	317	1.7
12	28	447	2.4
11	40	633	3.4
10	54	856	4.6
9	70	1118	6.
8	93	1435	7.8
7	120	1900	10.2
6	152	2460	13.2
5	198	3205	17.2
4	273	4380	23.5
3	396	6340	34.
2	635	10150	54.5
1	---	-----	---

METRIC CONVERSION TABLE (LIQUIDS)

Weight of Water

Weight of one cubic foot of pure water:

At 32° F.	=62.418 lbs.
At 39.1° F. (max. dens.)	=62.425 lbs.
At 62° F.	=62.355 lbs.
At 212° F.	=59.75 lbs.

CONVERTING SPECIFIC GRAVITY INTO DEGREES BAUME AND VICE VERSA

For liquids lighter than water:

$$°\text{Baume} = \frac{140}{\text{sp. gr. } 60°/60° \text{ F.}} - 130$$

$$\text{Sp. gr. } 60°/60° \text{ F.} = \frac{140}{130 + °\text{Baume}}$$

For liquids heavier than water:

$$°\text{Baume} = 145 - \frac{145}{\text{Sp. gr. } 60°/60° \text{ F.}}$$

$$\text{Sp. gr. } 60°/60° \text{ F.} = \frac{145}{145 - °\text{Baume}}$$

ORIGIN, SPECIFIC GRAVITY, ETC., OF OILS

Name	Type	Spec. Grav.	Viscosity	Color
Cylinder "A"	Mineral	894	146+	Brown black
Cylinder "Cold Test"	Mineral	886	116+	Reddish green
Castor Oil	Vegetable	963	---	White
Lard Oil	Animal	915	---	------------
Neatsfoot	Animal	913	---	------------
Olive Oil	Vegetable	916	---	------------
Pale Oil	Mineral	865	50	------------
Sperm Oil	Fish	875	---	White
Valve Oil (light)	Mineral	887	149+	Light yellow
Valve Oil (dark)	Mineral	887	152+	Greenish brown
Colza	Vegetable	914	---	White
Stearine	Animal	---	---	White gray

MEASUREMENT BASED ON U. S. GALLONS

1 U. S. Gallon = 231 cubic inches
= 0.133 cubic feet,
= 8.3356 pounds at 62° F.
1 cubic ft. = 7.48 U. S. Gallons.
1 Imperial Gal. = 1.2 U. S. Gallon.

Weight of a Cubic Foot of Water, English Standard, 62.321 pounds Avoirdupois.

LIQUID MEASUREMENT

1 cubic foot of water	= 62.3791 lbs.
1 cubic inch of water	= .03612 lbs.
1 gallon of water	= 8.338 lbs.
1 gallon of water	= 231. cubic in.
1 cubic foot of water	= 7.481 gallons.
1 pound of water	= 27.7 cubic in.
1 cubic meter	= 264 gallons.
1 Imperial gallon	= 1.2 gallons (U. S.)
1 acre foot	= 326,000 gallons.
1 pound	= .12 gallons.
1 pound pressure	= 2.31 ft. head.
1 atmosphere	= 34 ft. head
1 inch of mercury	= 1.134 ft. head
1 meter	= 3.281 ft. head.
1 cubic foot per second	= 449 G. P. M.
1,000,000 gallons per day	= 695 G. P. M.
1 miner's inch (California and Arizona)	= 11.2 G. P. M.
1 miner's inch (Utah, Idaho, Montana, Nevada, New Mexico, Oregon and Washington)	= 9 G. P. M.
1 miner's inch (Colorado)	= 11.7 G. P. M.
1 cubic meter per hour	= 4.22 G. P. M.
100 liters per hour	= .411 G. P. M.
1 liter per second	= 15.852 G. P. M.
1000 per hour	= 2 G. P. M.

TABLE EQUIVALENT FOR FUEL OIL

SPECIFIED DIESEL ENGINE FUEL OIL

When procuring Fuel Oil for Diesel Engines, an oil for fuel purposes should possess the following specifications:

Gravity	23 to 25 Baume
Specific Gravity	.917 to .905
Flash, Closed	.200 minimum
Viscosity	70 degrees Fahr. — 650 sec. max.
Sulphur	½ maximum

When "light distillate oil" is desired, the Gravity should be at its maximum 30.0 to 35.0 and 150 degree Fahr. min. Flash,

TABLE OF SOLID LIQUIDS, SHOWING TEMPERATURES

Temp. (Fahr.) Degrees	Distillate	Per Cent	Specific Gravity	Flash Point (Fahr.)
	Commercial Gasoline			
250-350	Kerosene, light	10	.73	50
160-250	Benzine, naphtha	10	.70	14
140-160	Gasoline, normal	2	.65	10
400	Kerosene, heavy	10	.89	270
350	Kerosene, medium	35	.80	150
482	Lubricating Oil	10	.905	315

NOTE: While there is a great variation, depending on values obtainable among the different petroleum products, nevertheless figures given on this table are the average.

DENSITY OF OIL

The following table gives the specific gravity and weight of oil corresponding to readings on Baume Scale:

Degree Baume	Specific Gravity	Pounds Per Gal.	Degree Baume	Specific Gravity	Pounds Per Gal.
12	.986	8.22	24	.913	7.61
14	.973	8.11	26	.901	7.51
16	.960	8.00	28	.890	7.42
18	.948	7.90	30	.880	7.33
20	.936	7.80	32	.869	7.24
22	.924	7.70			

HEAT OF VAPORIZATION

In the following table, the heat of vaporization is given of various liquids (at atmospheric pressure, except when otherwise specified):

Alcohol (ethyl)	208.92 cal.	
Ammonia (NH_3)	294.21	(at 7.8°)
Benzol	93.45	
Bromine	45.60	
Carbon dioxide	56.25	(at 0°)
Carbon disulphide	86.67	
Chloroform	58.49	
Ether (C_4H_{10})	91.11	
Iodine	23.95	
Mercury	62.00	
Sulphur dioxide	91.7	(at 0°)
Water	535.9	

CORRESPONDING VALUES OF SPECIFIC GRAVITY, DEGREES BAUME AND WEIGHT

Specific Gravity	Nearest Degree on the Baume Scale	Pounds per Gallon	Specific Gravity	Nearest Degree on the Baume Scale	Pounds per Gallon
0.70	70	5.84	0.86	33	7.17
0.71	67	5.92	0.87	31	7.25
0.72	65	6.00	0.88	29	7.34
0.73	62	6.09	0.89	27	7.42
0.74	59	6.17	0.90	26	7.50
0.75	57	6.25	0.91	24	7.58
0.76	54	6.34	0.92	22	7.67
0.77	52	6.42	0.93	21	7.75
0.78	50	6.50	0.94	19	7.84
0.79	47	6.59	0.95	17	7.92
0.80	45	6.67	0.96	16	8.00
0.81	43	6.75	0.97	14	8.08
0.82	41	6.84	0.98	13	8.17
0.83	39	6.92	0.99	11	8.25
0.84	37	7.00	1.00	10	8.34
0.85	35	7.09			

CALORIFIC VAULES OF THE PRINCIPAL CONSTITUENTS (LIQUID SUBSTANCES)

Liquids	Volumetric Expansion Centigrade	Volumetric Expansion Fahrenheit
Acid, nitric	.110	.061
Acid, sulphuric	.063	.035
Alcohol	.104	.058
Mercury	.018	.010
Oil, turpentine	.090	.050

DIFFERENCE IN FUEL CONSUMPTION ON HIGH ALTITUDES:

As the elevation above sea level increases, the pressure of the atmosphere and consequently the weight of the air drawn into the cylinder decreases. This in turn reduces the amount of fuel which can be consumed and thereby the power of the engine.

The reduction is about three and one-half per cent for each thousand feet (305 meters) elevation above sea level and occurs with all internal combustion engines. For example:

For an engine to operate at an elevation of 6,000 feet, the reduction would be .035 × 6, or .21. Thus an engine rated at 100 H. P. at sea level would be rated at 79 H. P. at 6,000 feet elevation. For altitudes under one thousand feet (305 meters), no reduction in the rating of the engine is made.

PETROLEUM SUBSTANCE.

The only natural liquid fuel is crude petroleum oil. This is distinctly a hydro-carbon liquid, and is found in abundance in certain localities in America and Europe, as well as some sections of Asia.

The principal sources of supply are, however, in the Ohio Valley of the United States, on the borders of the Caspian Sea in Eastern Europe, and Western Asia. It is found principally in porous sandstones, but also in natural cavities beneath the earth's surface, whence it is either pumped, or flows to the surface after the manner of operation of an artesian well.

Crude petroleum is dark brown in color, with a perceptible greenish tinge, and has a specific gravity which averages about 0.8. It is composed of a great number of liquid hydro-carbons, varying widely in specific gravity and chemical composition, and each seperable from the others by fractional distillation. The ultimate analysis of an average sample indicates about the following composition:

Carbon	84 per cent
Hydrogen	14 " "
Oxygen	2 " "
	100 per cent

Allowing for the combination of the inherent oxygen, with its equivalent of hydrogen to form water, the practical composition becomes:

Carbon	84 per cent
Hydrogen	13.75 " "
Water	2.25 " "
	100 per cent

The heat value of a pound of petroleum of the above composition is, therefore:

Carbon	$0.84 \times 14.650 = 12.306$ B. T. U.
Hydrogen	$0.1375 \times 62.100 = 8.539$ B. T. U.
	20.845 B. T. U.

PHYSICAL PROPERTIES OF OIL

Classification of oils according to their density is very commonly used to denote other characteristics. When alluding to "heavy" oils, we term it "viscous" and sluggish with a high percentage of asphalt and comparatively low heat value, while a light oil is supposed to be very "fluid" at ordinary temperatures, very volatile and rich in the lighter hydro-carbons and high in heat value. While in general, these characteristics hold true enough to explain the prevalent associations of ideas, there are so many exceptions and variations that it is essential to clearly specify the various properties of a particular oil in order to identify it.

EXPANSION OF WATER, MAXIMUM DENSITY = 1

C°	Volume	C°	Volume
0	1.000126	50	1.011877
4	1.000000	60	1.016954
10	1.000257	70	1.022384
20	1.001732	80	1.029003
30	1.004234	90	1.035829
40	1.007627	100	1.043116

COAL TARS AND COMPOSITION:

Water	Should not exceed 1 per cent.
Sulphur	Should be about .5 to 1 per cent.
Ash	Should not exceed 1 per cent. (Ingredients in unburnt quantities are harmless).
Pitch	Tar oils which contain a high percentage of residue beginning to vaporize at 400° C., the same results can be expected as with tar; in this instance the result will be the settling of considerable foreign matter in the engine with the consequential requirement of cleaning and grinding of exhaust valves.
Specific Gravity	Usually between 1.0 and 1.1.
Color	While tar oils generally show dark black color, nevertheless with the intermix of lighter ingredients a dark brown color is often found. Black residue signifies a large percentage of carbon or other heavy solving substances making up the composition of tar.
Flashpoint	Usually between 10° F. and 130° F.
Viscosity	On an average 2° Engler.

HEAT VALUES OF VARIOUS OILS

	Specific Gravity	Per lb. B. T. U.	Authority
California—Coalinga Field	0.927	17.177	Bashore
Bakersfield	0.992	18.257	Wade
Kern River	0.950	18.854	Bashore
Los Angeles	0.977	18.280	Bashore
Monte Christo	0.966	18.878	Bashore
Whittin	0.936	18.240	Wade
Texas—Beaumont	0.924	19.060	U. S. Navy
Beaumont	0.903	19.349	Bashore
Sabine	0.937	18.662	Bashore
Pennsylvania	0.886	19.210	Booth
Mexico	0.921	18.840	Bashore
Mexico	0.981	17.551	Bashore

CALORIFIC VALUES OF THE PRINCIPAL CONSTITUENTS OF FUELS:

The table below gives the calorific values of the principal constituents of fuels for Diesels. The values noted "at constant pressure" are alluding to such types of machinery where constant pressure is the factor to be considered. The values in this table are based on those determined by Berthelot, Thomson, and others.

Table of Calorific Values of the Principal Constituents of Fuels:

	At constant pressure		At constant volume	
Combustible	C. H. U.	B. T. U.	C. H. U.	B. T. U.
Hydrogen	34500	62100	34095	61371
Carbon burned to CO_2	8100	14580	8100	14580
Carbon burned to CO	2416	4349	2416	4349
Carbon monoxide (CO)	2436	4385	2426	4367
Methane (CH_4)	13344	24019	13276	23897
Ethylene (C_2H_4)	12182	21928	12143	21857
Sulphur	2300	4140	2300	4140

CHAPTER VI.

QUESTIONS AND ANSWERS ON DIESEL ENGINE OPERATION:

1. Give a Brief Definition of a Diesel Engine:

The Diesel engine is a machine which generates its motive power by the process of combustion. Its ignition system is compressed air. The burning of solid liquid sprayed in the cylinder creates a constant pressure, etc. etc.

2. How is the Diesel Engine Classified in Regards to Construction?

The two stroke cycle, commonly known as the two-cycle and the four-stroke cycle, known as the four-cycle.

3. Define the Meaning of Mechanical Efficiency of the Engine:

This applies to the ratio between the brake-horse power and actual power developed in the cylinder.

4. What is the Indicated Brake Thermal Efficiency?

The percentages of the heat units of the fuel that the engine is capable of transferring into indicated or effective work; or the ratio between the equivalent of the horse power in heat units and the number of heat units which the engine requires to develope one I. H. P. or one B. H. P.

5. Define the Volumetric Efficiency in a Four-cycle Engine:

The ratio between the weight of the air contained in the cylinder at the commencement of compression stroke and that required to fill the same volume with air at atmospheric pressure.

6. What is Meant by the Scavenging Efficiency in a Cycle?

The ratio between the weight of air in the cylinder at the commencement of the compression stroke and that of the mixture of air and burned gases.

7. What is Necessary to Maintain a Diesel Engine?

Fuel, Ignition, Water-Cooling and Lubrication.

8. Explain the Working Principle of a Four-cycle Diesel Engine.

During the first downward stroke the piston draws air through the suction valve; during the return stroke the suction valve and every other communication with the atmosphere is closed and the air in the cylinder is compressed. Toward the end of the stroke the fuel pump injects into the cylinder the quantity of oil necessary for the combustion stroke, so, that when the piston arrives on the dead center the

fuel burns rapidly, raising the temperature and the pressure in the cylinder. During the next downward stroke of the piston the burned gases are expanded, producing useful work. During the fourth stroke the piston sweeps out the burnt gases into the atmosphere through the open valve, after which the cycle recommences.

9. How Much Pressure is Necessary in Supplying Cylinders with Fuel Oil?

The pressure necessary for fuel injection varies with the load and the type of the engine, but is seldom lower than 540 pounds to the square inch, nor higher than 1005 pounds per square inch.

10. State the Process of Injecting Fuel in the Cylinders.

The pumping method is exclusively adopted. This owing to the fact, that a force-feed system is necessary exerting a pressure in excess with the existing pressure in the cylinder.

11. Explain the Function of the Fuel-injection Valve.

The fuel-injection valve is one of the characteristic parts of the Diesel engine. Its functions are two-fold: first, that of a valve to introduce the fuel oil into the cylinder at the correct moment; and, second, that of a sprayer to divide the fuel into minute particles.

12. What are the Duties of a Compressor on a Diesel Engine?

The duties of the Compressor on Diesel Engines are to supply high pressure necessary for the injection of the fuel oil into the working cylinders during the running of the engine, and to supply air for storage purposes into cylindrical steel reservoirs.

13. What are Reservoirs of Cylindrical Form Called and What are They Intended For?

They are called the starting bottles and act as a storage reserve power connected to cylinders through a system of valves and pipes.

14. How Much Air is Usually Required to Start Engine?

Depending on the size and respective type of engine.

15. How is a Compressor Constructed?

In two or three stages, between each of which the air is cooled by passing through a reservoir of water.

16. What advantages are there by Using Multiple Stage Compressors?

The multiplication of the stages of compression improves the volumetric efficiency of the compressor and diminishes the amount of work absorbed besides allowing of better cooling of the air.

17. What is a Scavenging Pump?

The function of the scavenging pump is to compress air to a low pressure to free the working cylinders of two-cycle engines of the exhaust gases, to be charged afresh with pure air for the next combustion stroke.

18. Can Scavenging be Effected Without Valves?

Yes, there is nothing to prevent the scavenging being effected without valves, as described for explosion engines, and applied in some Diesel engines, more especially for low powers. Scavenging with valves is more complete in its effect, and the weight of air which remains in the cylinders is greater. When the scavenging is carried out by means of single ports, the latter are closed before those of the exhaust, and so the pressure in the cylinder is no greater than that of the atmosphere. On the other hand, with valve scavenging the valves are closed after the piston has covered the exhaust ports, and so the pressure of the air in the cylinder before the compression stroke commences is that given by the scavenging pump, between 3 and 7 pounds per square inch.

19. Why is the Diesel Engine Classified as Constant Pressure Engine?

Mechanical means are adopted for obtaining pulverization or spraying by assistance of an injection device, the heavy fuel-oil into the cylinder by means of a current of air at a pressure considerable higher than that present in the cylinder itself; thus the fuel is subdivided into minute particles and forms a kind of mist. If the air in the cylinder at the instant when the injection takes place is at a sufficiently high temperature, the mist of oil spontaneously ignites, and the combustion lasts the whole time during which the oil continues to enter, assuming the character of gradual combustion as opposed to explosion. In this way the combustion takes place in "Constant Pressure" or Diesel Engines.

20. Why are Heavy Oil Engines Classified as Internal Combustion Engines?

Heavy oil engines convert the heat energy of the fuel into the engine cylinder itself. The heavy oil, injected into the cylinder in a suitable condition, ignites, burning with the oxygen of the air therein, and so evolves heat.

21. Why are a Great Many Engines Equipped With Only One or Two Air Starters?

As a rule, only one or two cylinders of a multi-cylinder engine are provided with starting valves, thus reducing the cost of the engine In some engines automatic means are provided for keeping the exhaust valves open during starting, compression being avoided until the engine is well up to or past its normal speed to save compressed air in starting. Some engines have a blow-off cock in the head of the cylinder, which is kept open for a time in starting; this is also used to blow off inimical substances.

22. Explain the Working of "Air-operated Piston Valves."

In some construction "mechanically operated" starting valves are eliminated, independent air-operated piston valves being substituted. These receive the starting air through a rotary distributor

operated by the cam-shaft. The rotary distributor can be connected or disconnected when the engine is running or is at rest. The fuel valve likewise is thrown in by a central control. This construction eliminates the cam and the fulcrumed lever for each starting valve used with the other types.

23. Explain the Functioning of "Actuating Valves."

The time of opening and closing of the different valves of Diesel engines must be accurately controlled; they are therefore not self-acting but are operated mechanically.

24. Explain the System of "Rocking Levers and Cam."

The usual method of actuating the valves on Diesel engines is through a system of rocker lever and cams; the latter are mounted on a horizontal shaft near the top of the cylinders. The cam shaft is driven through a set of helical gears, running in oil. by a vertical shaft which in turn is driven through another set of helical gears from the engine shaft. The vertical shaft has the same speed as the main shaft; the cam shaft runs at half the speed of the main shaft in engines having a four-stroke cycle and at the same speed as the main shaft in engines having a two-stroke cycle. The governor, usually of the through-shaft type, is mounted on the vertical shaft, as this has the higher speed. This arrangement permits the use of a smaller governor of the standard type.

25. How Are Pistons on Diesel Engines Water-cooled?

The water for piston cooling may be circulated through telescopic pipes, with the stuffing box a moving part of the piston, or the stuffing box may be attached to the frame and the water be supplied to the piston through hollow walking arms through which the water flows through the pipes leading to the piston. A pump may be actuated from the crosshead and the water be carried to the piston through the hollow piston rod. In another cooling system water is sprayed by air against the heated piston surface, not enough water being used to fill the water space of the piston. The excess water drains off through a pipe surrounding the spray pipe, no stuffing box being used.

26. Is the Trunk-piston Preferrable to the Cross-head?

The trunk piston is more easily provided with a greater bearing surface than a cross-head; thus insuring less wear. The lubrication under pressure of a cylindrical guiding surface is more effective than with open guides. The piston moves over perfectly cooled walls, whereas the cross-heads tend to heat more readily and when once hot is not easily cooled; moreover, water-cooled cross-heads complicate construction. For large engines this construction is used, as it affords greater accessibility and ease of adjustment, the cross-head guides in such engines being water-cooled.

27. What Are the Usual Methods in Lubricating Cylinders on Diesel Engines?

The cylinders are lubricated by providing each cylinder with a force-feed pump operated by a reducing motion from the end of the piston or with a multiple-plunger pump driven by an eccentric device direct from the cam shaft, one plunger being provided for each working cylinder and one for each air-compressor cylinder. In some engines two plungers per cylinder are provided, one for each pair of the four oil feeds grouped around the cylinder. Two-stroke Diesel engines usually have separate oil feed above and below the exhaust ports to prevent any excess of lubricating oil from being swept through the exhaust ports.

28. Explain the Retarding Method of Injecting Air and Fuel.

In the cylinder of large engines the fuel needle can be so governed that the entrance of the injection air and fuel into the combustion space is retarded and gradual, preventing excessive use of injection air cooling of the atmosphere at the point when the ignition must be maintained. By these means tar oil can be ignited without the use of ignition oil. To remove the fuel needle from the valve it is necessary in some constructions to remove the nut holding the spring bonnet and cap into place, as well as the two nuts that fix the position of the rocker arm through which the needle valve is actuated.

Construction that facilitates the removal of the fuel needle is of decided advantage, as the valve needle has to be removed when it has to be ground and frequent grinding is necessary. Fuel needles have to be ground weekly or monthly, depending on the quality of the fuel oil. The use of oils with high ash contents necessitates more frequent grinding of the needle. The needle is turned back and forth in its seat, a small quantity of emery dust and oil should be used.

29. Explain the Construction of Valve Attachment.

Connections of the copper seamless steel tubing used for carrying the air and fuel under high pressure are made with joining the ends of the pipes to a copper shank terminating in a cone which fits into a tapered seat machined out of the body. A steel gland nut slipped over the copper shank is pressed against the cone to seal the seated connection. To seal the air chamber of the fuel valve and prevent leakage of air around the valve needle, lead or babbit metal shavings mixed with flaked graphite are used as packing material, secured by appropriate glands, a series of labyrinth grooves on the valve needle constitutes an added precaution against serious leaks.

30. At What Degree does Timing Usually Occur?

The time at which the different valves are opened or closed differs widely with different makes of engines. The air valve opens 15 degrees to 20 degrees before the piston reaches the top dead center, and closes 15 to 20 degrees past the bottom center, being open

a total period of 210 to 220 degrees. The fuel valves open 2 degrees to 8 degrees before the piston reaches the top center, and closes 18 to 36 degrees after the piston has passed the top center, being open 20 degrees to 44 degrees. The exhaust valves open 25 to 45 degrees before the piston reaches the bottom center, and closes 8 to 14 degrees after the piston has passed the top center.

It should be taken in consideration, that the type of fuel valve and the properties of the liquid fuel burned, greatly influences the timing of the fuel valves.

CAUSES AND EFFECTS IN THE PRINCIPLE OF OPERATION OF DIESEL ENGINES AND REMEDIES.

1. What is the Maximum Piston Travel Per Minute and R. P. M.

Similar to the Steam Reciprocating Engine, the Diesel Engine is rated in piston travel to 1500 feet per minute. Its revolutions per minute are rated with rare exceptions up to 400 R. P. M.

2. How Should Valves Be Set on Diesel Engines?

To set the spray, intake and exhaust valve, first know that shafts are in their proper position. Adjust cam roller clearances; giving the exhaust about .30 degree, intake about .025 degree and spray about .018 degree.

Jack engine ahead until the pointer on the after end of the housing is in line with some graduation on the fly-wheel that is marked for the opening of some valve. When the engine is in the position where some valve should open, the cam roller for that valve should be just starting on the cam; and if not, adjust the roller by screwing in or out on adjusting screw in upper end of rocker arm. Proceed on all other valves the same as the first. After setting of all valves go over roller clearance once more. After final adjustment no roller should be left on the cam or close enough that the expansion of the valve stem will cause the roller to ride the cam. This applies specially to the exhaust, because that valve gets more heat than any of the others. All relief valves are set by connecting them to a hydraulic testing machine and adjusted to open at their required pressure.

(Note: Above rule for setting valves is identical with almost every type of Diesel Engine, differing in minor details).

3. What May Cause the Engine to Slow Down?

Water in fuel, piston seizure, hot bearings, propeller (on marine) fouled, low compression,—always, if mechanical defects are not suspected, test fuel oil, and if the fuel is up to standard requirement the engine should be stopped and again throughly examined.

4. What Precaution Should be Taken With Circulating Water and Spray Air System?

Strainers should be kept clean. All valves kept in good repairs. All leaks should be stopped immediately after they are located.

5. What Causes Non-circulating of Water Through Engine? How Detect? How overcome?

Air pockets in water-space or foreign matter blocking up passages. If there are air-pockets, they can be detected by hot spots in cylinders. Usually the air-pockets shift from space to space and can be eliminated by opening air-cocks on cylinders.

6. What May Cause the Air Compressor to Furnish Insufficient Air? How is it Indicated? How Locate Trouble? What to Do?

Leaky valves, leaky rings, too much clearance on top of pistons. If suction valves are leaking, the suction line will heat and gauge pressure on proceeding stage will run high. If discharge valves are leaking, gauge pressure on proceeding stage will run low. Leaky rings will cause the first stage to build up; if it is the second or third stage, or if the first stage, air will blow into crank case.

If volume clearance is too much, all pressure will run high except the third stage. If valves are leaking, they must be re-ground. If rings are leaking, they must be re-newed. If clearance is too great piston must be raised.

7. What Causes Cylinder and Cylinder-heads to Crack? How to Detect?

One of the principal causes is over-heating. In the case one cracked, it will be observed by lack of firing, causing stopping. To detect, shut off fuel and open try-cock, when water will be noticed flooding cylinder.

8. If No. 5 Cylinder Head is Cracked so that the Circulating Water Runs into the Cylinder; it is Important that the Engine Continue Running. What to Do.

Cut a strip of tin that is wide enough to cover water port in the head and narrow enough to pass between the studs that secures the flange to the head. Loosen off nuts on studs and force the tin between the flange and the gasket, leaving the gasket between the tin and the flange on the connection to the header. Secure the nuts and close off the valve that connects water line to the bottom of that cylinder. Free inside of cylinder of water. Drain water out of the head. Cut out that cylinder, so that it cannot fire and run again.

9. The Exhaust Rocker Arm Breaks On No. 7 Cylinder of Starboard Engine, (when twin) there are no Spare Rocker Arms Aboard. What to Do, to Keep Engine Running.

Remove broken parts. Take the intake rocker arm and put it in place of broken rocker. The intake valve will open with the suction created by the piston on its suction stroke. If the valve spring is too

strong, same may be ground off of each end. This cylinder will not get enough air for full power but will do very well for normal running.

10. How Would You Adjust the Compression on the Working Cylinders?

First, take compression cards on all working cylinders; which is done in following manner:

Connect Indicator gear to the cylinder. Start circulating pump. Build up pressure.

Turn the engine over (with motor, when equipped with same) and build up speed to about normal running speed. Then open the cock allowing the pencil of indicator to make four or five strokes on the card. Then close off cock and shut down the engine. Take a card in this way on each cylinder of engine. When taking cards the following should be noted: There should be no valve leaks. Roller clearances should be adjusted and then the engines should be run at the same speed for each cylinder when taking cards.

After cards are taken, the cards must be carefully measured to find out how much compression there is in each cylinder; then the compression can be raised or lowered as the case may require in the following manner:

Jack the engine over, so piston reaches top center of cylinder to be worked first. Remove the nuts from the crank pin bolts, leaving bolts and brasses in place . The bolts can be locked in place. Put pin through bottom cylinder wall beneath the piston to take the weight of piston.

Again jack the engine slowly by hand until crank has traveled far enough on its own down stroke, to allow the piston to come down on the pin through the cylinder. The pin will hold the piston and allow the crank brass to leave the connecting rod. When doing this, the crank pin bolts must be carefully watched, so that they will not catch in the holes in the foot of the rod. After the brass is far enough from the rod, the compressor liners can be attended to also; minimizing or adding as the case may require.

After the right amount of liners are between the top of the brass and bottom of the rod, jack the engine back to top, secure the bearing and remove the pin which is through the bottom of the cylinder

When working on the cylinder is completed, compression must be taken again and cards measured as before. Each cylinder should have very near the same compression, which is usually about 480 pounds per square inch, although cylinders will fire with much less compression.

It is advisable, to not allow over 10 pounds difference between the highest and lowest compression to exist.

When adjusting compression after installing a new piston or new rings, on one piston it should be set about 10 pounds lower than the other cylinders, because as the rings wearing in, the compression

will increase until the rings are worn smooth, then compression should be readjusted. It will usually be found too high.

11. How is the Governor Adjusted?

Usually on Diesel governors, the place where the governor cuts off the fuel, is controlled by the tension on the top spring in the governor. The tension of the spring is regulated by a threaded bushing at the lower end of the spring. The bushing is secured with a lock nut. The greater the tension on the spring the greater speed the engine will reach before governor cuts off the fuel. The adjusting bushing is screwed in a certain distance, thereby bringing a certain tension to bear on the spring.

The engine is then run to ascertain where the governor cuts off the fuel. The tension may then be properly regulated according to the required revolution. The lower spring only takes up all lost motion, keeping the stem from oscillating.

12. If Engine Should Lose Compression, Where Would You Find the Trouble?

(1) Sticky inlet valve; (2) Pitted or corroded exhaust valve; (3) Improper seating of either valve; (4) A loose or open compression tap; (5) Defected piston rings; (6) Leaky gaskets; (7) Lack of lubrication around cylinder walls, etc.

13. How Would You Detect a Leaky Spray Valve?

There will be a loud thump in the cylinder, caused by premature firing.

14. How Would You Detect a Leaky Fuel Check?

Cylinder will fire light and some times miss, especially when running slow or when carrying high spray air. To test: Open fuel by-pass. If air blows through the drain line, it is a sure sign of leaky check.

15. How Would You Detect Too Low Spray Pressure?

Spray valve will heat; caused by fuel burning in the valve body. (This is liable to cause sticking of valve) If the spray pressure is high enough to prevent oil burning before it enters the cylinder and jet not high enough to give a proper mixture, there will be black exhaust, caused by incomplete combustion.

16. How Would You Detect Excessive Lubrication?

White smoke, often is caused by too much lubricating oil burning in the cylinder.

17. How Would You Detect Improper Combustion?

Black smoke denote improper combustion.

18. How Would You Detect a Leaky Exhaust Valve?

A leaky exhaust valve will usually heat up and sometimes stick. The temperature of the exhaust gases will be too high.

19. How Would You Detect One Cylinder Missing?

The engine will slow down, and if floating the motor, (where motors are in use for starting purpose) the Ampere-meter will register an irregular load, and by opening try cocks.

20. How Would You Detect Too Low Compression?

Cylinders are liable to miss, especially if running slow. Use indicator and take compression card by cutting off spray air and opening on fuel, so there will be no expansion and no raise in pressure caused by injection air.

21. How Would You Detect Leaky Air Cooler?

The discharge pipe from cooler to next stage suction will heat up; caused by air blowing the water from around the coil. This will frequently happen if the next stage suction is leaking, but if the valve is leaking, the end of the pipe next to compressor will be hottest. The gauges will also show if the valve is leaking.

22. How Would You Discover if the Third Stage Coil Is Leaking?

If the third stage coil is leaking, the spray air line will heat up and very often the entire engine will heat or the cylinder next to the leaky coil; or if the engine has a relief valve on the circulating water it may raise.

23. How Would You Detect a Leaky Oil Cooler?

This can be discovered, if the water pressure is greater than the oil pressure, accomvanied by filling up sump tank, and if tested with nytrate of silver it will show wsalt. If the oil pressure carried is greater than the water pressure, the pump will run down too fast with a consequential greasy circulating water discharge.

24. How Would You Detect a Leaky Air Starting Valve?

This can be discovered by extreme heating of pipe leading towards starter.

25. How Would You Detect a Sticky Spray Valve?

Sticky spray valves will cause a loud thump in the cylinder and sometimes raise the cylinder relief valve.

26. How Would You Detect Cam Rollers Being Heated?

This may be caused by the expansion of valve stem, causing the roller to be in contact with the cam all the time. They may also hold the valve off its seat.

27. If the Third Stage Air Compressor Discharge Valve was Acting Slow, Where Would You Find the Cause?

This may be caused by broken spring or carbon on valve. This will cause the second stage pressure to run too high.

28. What Causes Heating of Clutch Collar?

This is due to spider not being pulled past center of the shoes. The collar should run free at all times.

29. What Causes Slipping of Clutch?

This is indicated by engine racing, also the clutch band on the fly wheel will heat up.

30. If Repeated Breakage of Crankshaft Should Occur, How Would You Account For It?

Repeated breakage of crank shaft of Diesel engine is likely due to unequal working in cylinders, causing shocks and undue impacts. A crank shaft usually breaks after the material has become crystallized, and when a break has occurred it may be taken for granted, that more or less crystallization has taken place throughout the whole crank shaft material. The original can be nearly recovered by heat treatment, and the whole crankshaft should be so treated occasionally, at least when ever part is repaired by welding.

31. What Effect Has An Undue Amount of Water in Fuel Oil of Engine?

Water lowers the heating value of the oil, as its evaporation consumes fuel; it also lowers the temperatures of the combustion space. A purchaser should not pay for water and the cost of transporting it when he is buying oil. Fuel oil should not carry more than 0.5 per cent of water. Mechanically entrained water, which will separate from the oil and accumulate in the bottom of the fuel tank, will cause failure of ignition, and if it displaces the oil in the fuel valves long enough, the engine will stop.

32. What is the Burning Point of Oil?

The "Burning Point" of an oil is the temperature at which it ignites and continues to burn in an open cup. The burning point is no criterion of the usefulness of an oil for use in Diesel engine, save that it is a further index of the fire hazard. The nearer the burning point to the flash point, if the flash point is low, the greater is the fire hazard. A low flash point and a high burning point indicate the presence of high volatile oils mixed with heavy oils. The burning point is 10 degree to 50 degree C. and rarely 100 degree C. higher than the flash point. Extreme differences usually indicate crude oils that have not been "topped," or mixtures of volatile and heavy residual oils. If the flash point is sufficiently high to preclude fire hazards, determination of the burning point is superfluous.

33. What Effect Will "Ash" Deposites of Fuel Oil Have on the Engine?

Ash is the most detrimental remnant of the burning of fuel oils for Diesel engines, as it causes excessive wear of cylinders and exhaust valves. The ash is usually composed of mineral particles of great hardness, such as quartz and sillicates, or oxides or iron aluminum, which, becoming mixed with the film or lubricating oils, adhere to the

piston and cylinder walls, accumulate, and causes excessive wear. An ash content in excess of 0.05 per cent will render an otherwise excellent fuel unsuitable for use in a Diesel engine.

34. What Importance Has Parafine Content in Fuel Oil?

The parafine content of an oil is important merely in its physical effect on the oil. Oil with an appreciable parafine content may solidify or become highly viscous at low temperatures (0 degree to 15 degree C.) Moderate heating of oils will obviate any difficulty from the sluggishness at low temperature due to parafine content.

35. What Effect Has "Asphalt" Content in Fuel Oil on Diesel Engines?

High asphalt content points to high content of constituents that will produce a coke residue. If the coke residue of an oil is satisfactory, its asphalt content may be disregarded as being of no importance. High asphaltum content makes an oil objectionable for use in engines with fuel-valves that are closed by fuel needles. It tends to gum the needle and causes it to stick. By heating the oil sufficiently to make it more liquid, this objection is greatly removed. Such oil is as a rule highly viscous and they have to be heated to cause the necessary fluidity.

36. If Engines Are Running and Bilges Are Afire What Should Be Done?

Stop engines immediately, give fire alarm and shut down blowers or any other machinery that causes circulating of air.

Start fighting fire with apparatus with which boat is provided. If fire has started from electrical appliances, all switches must be pulled. If fire cannot be stopped any other way, it may be smothered by getting all hands out of the engine room and close water-tight doors and prevent all fresh air from entering engine room.

MACHINERY MATERIAL

1. Define the Meaning of Strain:

Whenever a force is applied to any member of a machine or structure the shape is altered. The change in any linear dimension is called the strain and the change per unit of linear dimension is called the unit strain.

2. Define Stress:

If a machine part is acted on by forces, there exists a tendency within a body to resist the external force to tear apart or to crush the body. The unit stresses and unit strains correspond respectively to loads applied to machine.

3. Ultimate Strength:

In tension tests of machine material there is usually a corresponding figure dealing with defined load of maximum proportion before rupture

should occur. The unit stress corresponding to this load is called the ultimate strength, or, more briefly, the ultimate for the material.

4. What Are the Principal Ingredients in Steel?

Carbon and iron are the principal ingredients in steel, but special steels of great strengths and toughness are made by alloying carbon and iron with other elements. Commonly alloy steels are: Nickel steel, Tungsten steel, Vanadium steel and Manganese steel.

5. Name the Uses for Following Metals on Machinery:

(a) Lead, Tin, Zinc: These metals are used in special cases in which strength is not requisite, and in which resistance to chemical action is necessary. They are used in alloys for bearings on machinery.

(b) Brass: This is an alloy for zinc and copper. It is used for small machine parts, in which resistance to corrosion is of importance. It is also used as a bearing metal.

(c) Bronze: This is an alloy of copper and tin. It is used where resistance to corrosion is necessary, and where strength is also required. It is used as a bearing metal in the highest grade of bearings. It is very expensive as compared with other metals of similar use.

6 What is the Approximate Composition of Babbit Metal?

The approrimate composition of babbitt is tin, 89 parts copper, 4 parts; antimony, 7 parts.

7. What materials Are Machine Frames Made Of?

For engines, where little vibration exists, cast iron is the material generally used on account of its low cost.

8. What Material Is Used for the Manufacture of Cylinders?

Cylinders are almost exclusively made of cast iron, while in lighter machines aluminum is used in particular for high speed engines. Usually the thickness is determined not by consideration of strength, but by comsideration of foundry practice. It is impossible to cast a very thin cylinder. Cast iron makes a much better bearing metal for the rubbing of the piston than does steel.

9. What Material Is Used for the Manufacture of Shafting?

Cold rolled steel is very widely used on account of the ease and cheapness with which it can be rolled true to shape and size.

10. Where Is Nickel Steel Preferable to Common Steel?

Nickel steel is widely used where high strength is necessary, as on valves on Internal Combustion Engines. A Nickel content of 3.5 per cent makes steel stronger and resistant to shocks. Nickel strengthens steel without reducing its ductillity to any great ertent, hence Nickel steel is tough.

11. What Effect Has the Adding of Vanadium to Steel?

Vanadium in the form of ferrovanadium adds great strength to steel,

and seems to be especially valuable in adding resisting power against repeated application of stress. Vanadium seems to benefit cast iron, probably on account of its tendency to remove oxygen from the iron.

12. What Effect Has the Adding of Tungsten and Molybdenum in Steel?

The presence of Tungsten and Molybdenum in steel so affects the critical temperature, at which steel changes from a very hard material to a much softer material, that with proper heat treatment Tungsten and Molybednum steels retain their hardness at a red heat.

13. What Effect Will Titanium Have on Steel?

Titanium is used as an ingredient for steel, and renders the steel more uniform in quality throughout.

14. What Effect Will Manganese Have on Steel?

A Manganese content of greater than 7 per cent makes steel very strong and tough, but so hard as to be practically unworkable. Manganese steel usually contains about 12 per cent of Manganese, and while by the exercise of great care it may be forged or rolled, it is usually cast directly into the desired shape of the finished product.

15. What Effect Will Chromium Have on Steel?

Chromium makes possible a steel of great hardness and strength. Very often it is used in connection with nickel in making special grades of steel.

16. What Effect Will Copper Have on Steel?

The adding of about 1 per cent of copper has no marked effect on the strength or ductility of steel, but greatly diminishes the tendency to corrosion.

17. What Effect Will Silicon Have on Steel?

A Silicon content of about 4 per cent increases the magnetic permeability of steel and also its electrical resistance. This combination makes an excellent steel for the magnetic circuits of electrical machinery. Silicon always tends to give steel an acid reaction, and hence a low Silicon content is always found in basic steel.

18. What Effect Will Carbon Have on Steel?

Carbon up to 1.25 per cent increases the strength of iron, and the increase is approximately proportional to the carbon content.

19. What Is Semi-Steel?

The melting of 20 to 50 per cent of steel scrap with pig iron produces semi-steel. The product is a cast iron of high strength and low carbon content; it is not steel. In the manufacture of iron castings in which strength is important, Semi-Steel is used.

CHAPTER VII.

FUEL FEED AND IGNITION

There is no doubt that maximum fuel economy is attained in a Diesel engine at a certain load when air and fuel are used in a certain proportion. But this point may be prevented by overheating, pounding, carbonization, or undue strain on the machinery. Engines designed for high compression are also designed for considerable excess of air at rated full load. Reducing this excess by injecting more fuel might shock certain mechanical parts beyond endurance. Too much air results in a lower temperature of ignition and, apparently, a slower rate of burning. Either conditions lowers efficiency for reasons pointed out in following lines, considering basic principles of Diesels.

The theoretical thermal efficiency of an internal combustion engine is given by the expression $(T_2 - T_1) = T_2$.

T_2 is the absolute temperature at the beginning of the working stroke and T_1 is the absolute temperature at the end of the same stroke. Absolute zero is 461 degrees below zero on the Fahrenheit scale. A temperature of 70° F., for instance, means 531° referred to absolute zero. From the above efficiency formula it is evident that a high temperature of combustion at dead center and expansion to low temperature are the theoretical as well as some of the practical requisites of fuel economy. High temperature at dead center is secured by high compression in a small clearance space and by complete combustion of just the right fuel proportion before the piston has begun the working stroke. If only part of the normal charge is compressed in a fixed clearance, or if the excess of air is too great, or if combustion continues during the working stroke, maximum efficiency cannot be obtained. No one engine in practical operation overcomes all these "ifs" under light load condition.

One theoretical way of securing the ideal condition would be (1) to vary the quantity of mixture containing constant proportions of fuel and air, and (2) to vary the clearance volume in accordance with the quantity of fuel mixture so as to maintain the same degree of compression at all loads. The first requirement is practically reached by the carburetion engine, but a fixed unvariable clearance volume exists in all types of engines treated herein. Consequently the carburetion charge does not receive maximum compression during light load. Since the air intake is not throttled down in semi-Diesel and Diesel cylinders at light loads, compression remains more nearly constant. But the proportion of fuel is cut down and combustion is supposed to be slower, both of which conditions reduce the maximum temperature. In one

respect the Diesel cycle is a cross between carburetion and semi-Diesel principles. Compression pressure is nearly constant as in the semi-Diesel cylinder but the total supply of air can be partly reduced with less fuel at light load (as in the carburetion cylinder) by governing the supply of injected air through hand regulation of compressor valves.

Prolongation of Diesel fuel injection avoids abortive pressures but prevents full utilization of heat from the beginning of the working stroke. Injection of water with fuel has a similar effect in other engines.

One valuable but only partial compensation, in the case of low compression in any type of engine, is the fact that a greater degree of expansion is secured at light load. Theoretically, appreciable fuel economy would be added by increasing the ratio of expansion beyond that existing in practical engines. A longer stroke relative to clearance would be necessary. The atmospheric volume of air would then have to be reduced for the beginning of compression to avoid injurious pressure at the end of this stroke. Design of expansion in practical machines is a compromise between fuel economy on one hand and commercial and mechanical limitations on the other hand. In general, the higher the compression in a given type, the lower is the ratio of expansion required to attain equal efficiency.

In the case of the semi-Diesel engine the supply of air for each working stroke at different loads remains nearly the same. This means a great excess of air at light load when the fuel supply is cut down. Compression pressure remains about the same, but explosion temperature is reduced. Because of the short time available for the fuel and air to mix in this type, it is probable that a larger excess of air is required. Also an efficient device is necessary to divide the oil into fine particles, especially when a heavy grade is burned. If the oil is not properly divided and distributed, it is liable to "crack" or decompose into parts from hydro-carbon gases down to pure carbon. This latter is the cause of much grief. In most semi-Diesel engines, oil is forced through one or more fine openings in the fuel nozzle. Its mechanical motion in contact with the hot ignitor surface help to mix it with the air. Some combustion chambers have practically no obstruction of openings into the cylinder. As will be seen in the article on the Fairbanks-Morse "Y" engine, which uses no injection water, represents the other extreme. Oil is injected against a hot tube fixed inside a spherical chamber. The bulb connects with the cylinder through a round opening.

Oil is broken up mechanically, in the Diesel cylinder, by the compressed air which accompanies it past the nose of the fuel needle. Excess of air is governed partly by its supply pressure, which in turn is controlled by hand valves on the compressor. The excess runs somewhat the same as with semi-Diesel operation. Combustion takes place directly behind the piston, but does not result in so sudden an explosion and shock, on account of the gradual admittance of fuel. The high

heat of compression effects positive ignition upon comparatively heavy oils.

Generally speaking, the higher the degree of compression just before ignition the greater is the efficiency. This is an example of the law applied to heat engines which states that the per cent of heat converted into work varies with the range in temperature during expansion. High compression in a small clearance volume results in a high temperature and pressure during combustion becoming available during expansion to lower temperatures and pressures.

Semi-Diesel and Diesel engines are allowed higher compression because fuel is not admitted until the piston is ready for the combustion impulse. Diesel type engines are of course built to withstand the highest pressures of all. Fuel is introduced by a further supply of compressed air during the first part of the working stroke, an arrangement which allows more gradual combustion instead of instantaneous explosion. The average period of time opening of the fuel valve may be considered from 12% to 15% of the stroke.

Experiments in efficiency tests of Diesels have proven, that the conversion of energy is well near to 35% from the fuel into outside mechanical work. From this maximum with various engines and loads, the percentage ranges down to zero, in which latter case just enough fuel is admitted to run the engine without load.

FUEL INJECTION VALVES

Of the numerous mechanical contrivances and different accessories necessary, none is more important than the device causing the fuel to be brought in direct communication with the existing heat in the cylinder, caused by compression, than the Fuel Injection Valve.

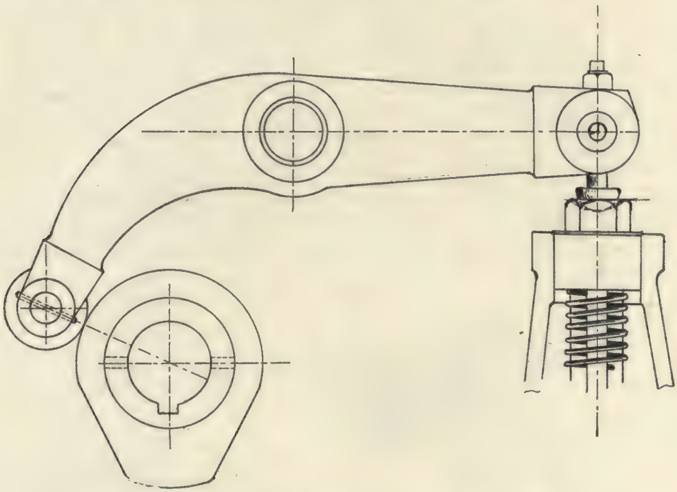
The functions the fuel injection valve has to perform are two-fold: In the first place, that of a valve to introduce the fuel oil into the cylinder with equal regularity; and the second, that of a sprayer, to divide the fuel into minute particles, causing a "constant volume."

The operation of this valve may be explained in following: The valve arrangement is formed by a needle, ending in a cone, held accurate on a conical seating by the valve spring. A lever actuated by a cam, raises the needle at the required moment, establishing communication between the fuel injection valve casing and the cylinder.

The spraying is effected by means of a number of washers, provided with small holes, and by a cone grooved along its gentrating lines, threaded by a tube usually made of bronze. In placing the washers, great care should be taken in properly "lining of holes on washers," the same never to be placed hole to hole, but rather so that the hole of the next following washer is irregular placed.

The washers may be minimized when a change of fuel takes place. That applies to difference in gravities, with consequental results in lower or higher viscosity, as the case may be.

A small hole, running through the center of the steel diaphragm directs the air, assisting the action of the valve in the proper distribution of its fuel.



Demonstration of actuating valves through cams

Very often, the clogging of the channel hole in the check valve causes complication and when taking the valve apart, the trouble may be discovered.

It is necessary to clean interior parts of fuel injection valve from foreign matters, and impurities assembling in the channel must be eliminated.

The oil delivered under pressure from the fuel injection pump, is directed into the fuel injection valve just above the perforated washers, and assisting the high pressure compressed air from the fuel injection bottles fills the sphere around the sleeve. At the moment the valve spindle raises, the air at 50 to 70 atmospheres—700 to 1,000 pounds per square inch—rushes into the cylinder, in which the pressure is now 30 to 35 atmospheres—430 to 500 pounds per square inch—drawing with it the fuel, which, in passing through the holes of the washers, is divided in a fine mist.

The fuel oil in this state of "fog" enters the combustion chamber, containing the heat of ignition, to use the expression, at the end of the compression strokes it spontaneously ignites. The combustion takes place practically at constant pressure throughout this period of the stroke, during which the oil continues to be forced into the cylinder.

It is vital that the valve actuating gear in functioning its lifting

and opening of valve should not be influenced by variation of load of engine. The pressure per ratio between the interior of the fuel injection valve casing and the cylinder should remain constant. In this

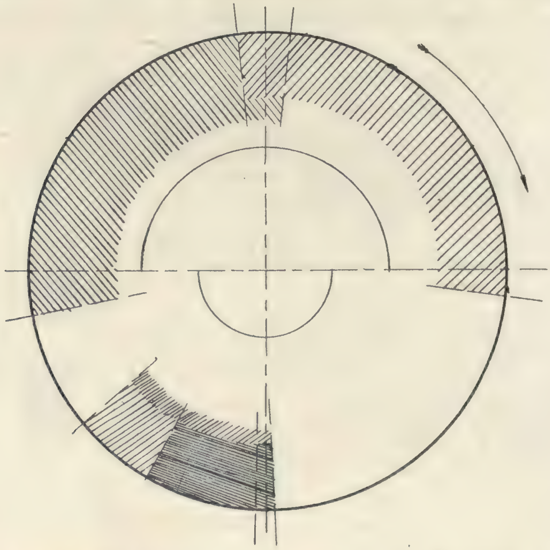
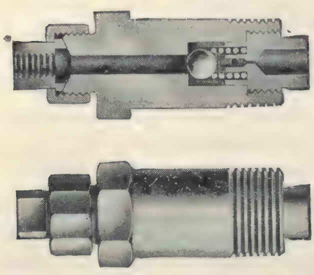


Diagram of Valve Settings of Crankshaft on Four-cycle Engine

way no variation of velocity efflux and the quantity of air issued with each working stroke occurs, immaterial of existing horse power of engine.

While the supply and the demand, regulated, corresponding to the load capacity of the engine, is taken care of through the fuel injection pump and its governor arrangement, it is seen that the desired amount of fuel necessary to keep up the momentum of impulse is in this way properly maintained.



Oil Injection Nozzle of the Ball-Check Type

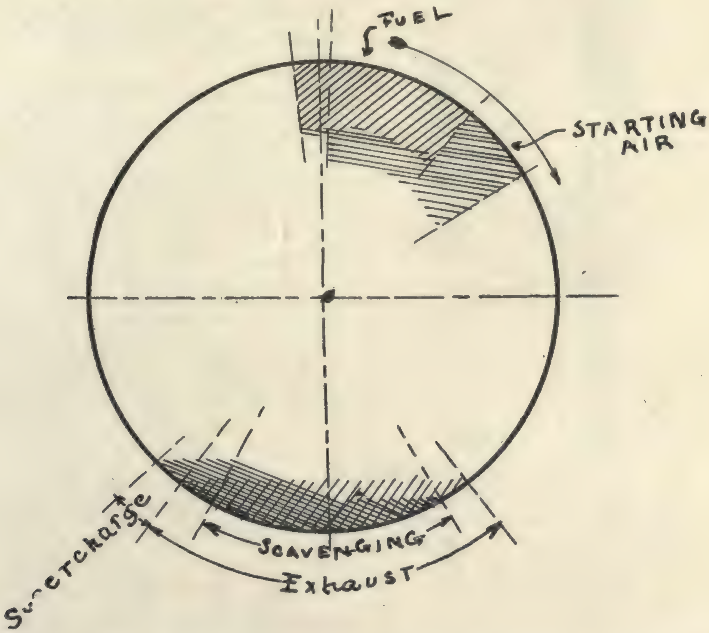
An oil injection nozzle used on the Giant Oil Engine is seen in the illustration. This is a distinctly exclusive feature patented by the Chicago Pneumatic Tool Company.

It is screwed directly into the center of the combustion chamber head. It contains a ball check valve and is a novel departure from the usual spring type and numerous similar injection nozzles used, working identical in working operation.

As the inspiration and compression strokes are common to all types of engines and the method of injection is the

main feature under discussion, a detailed description of the injection and combustion period will be gone into.

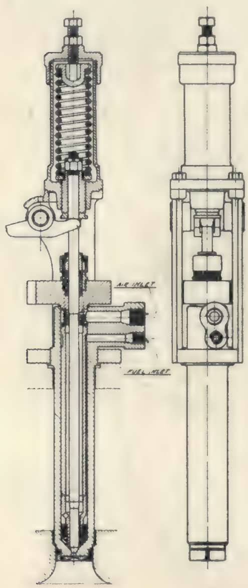
During the inspiration stroke, a measured quantity of fuel is delivered near the bottom of the fuel injector just above the needle valve on engines using the method of established types generally used. The injection air, which in all cases is well above the compression pressure, is forced against the fuel, atomizes it and forces it into the cylinder. One point worthy of note is the fact that the action of the injection air thoroughly atomizes the fuel prior to injection, and upon this atomization depends the combustion efficiency of the cycle. The amount of fuel, as before stated, is regulated by the fuel pump and governor. The duration of the injection or combustion is also regulated according to the load on the engine. For light loads the injection valve remains open a shorter period than for heavy loads. Aside from the advantage of thorough atomization there is one great objection to this method of injection, namely refrigeration during injection.



Valve Settings of Double-Port-Scavenging Two-cycle Engine

As the compressor pressure is about 500 pounds and the injection pressure between 700 and 1000 pounds, there will be a very rapid expansion of the injection air from the maximum to the compression pressure. This rapid expansion causes a great reduction in the temperature of the injection air and fuel, just as rapid compression causes

an increase in the temperature. The effect of this sudden reduction of temperature causes a time lag in the combustion—because cold fuel is difficult to ignite—as well as the collection of small particles of fuel on the piston, at which point local burning occurs. The result of this is an invariable sag of the piston head due to excessive local temperature. Were it possible to use highly heated air for injection, this refrigeration would to a great extent be overcome. However, with the Diesel method of injection, combustion would take place to the injection valve prior to the injection period. It is well to note, that in the Diesel engine the fuel injection is in mechanically timed relation to the piston position, but as the injection and compression pressure are always constant, the rate of injection is also constant.



Carels Type of Fuel Inlet Valve used on Nordberg Diesels

In accompanying illustration a Carels type of fuel inlet valve is shown. It follows the principle of valves used on most engines. The amount of oil desired to enter is automatically regulated by the action of the governor on the pump, depending on the requirement.

In accomplishing the pulverizing the oil is forced through its entire passage into the spraying arrangement, consisting of four metal rings with usually about 20 holes drilled into them of the size of 1/10 to 1/16 of an inch in diameter. The holes in the washers, or rings, are placed in staggering position to accomplish proper spraying results. Beneath the washers is a conical shaped piece, acting as a guide to allow the oil to pass into small passage ways, in similarity to nozzles.

It enters then direct into the cylinder by the expanding orifice, made of steel, the guides of the needle valve usually cast of iron. With the high pressure of the injection air, in the period when the lifting of the needle valve takes place, the fuel is forced through the pulverizer by the air in minute particles of fine spray into the combustion chamber where it is ignited coming in contact with the compression temperature.

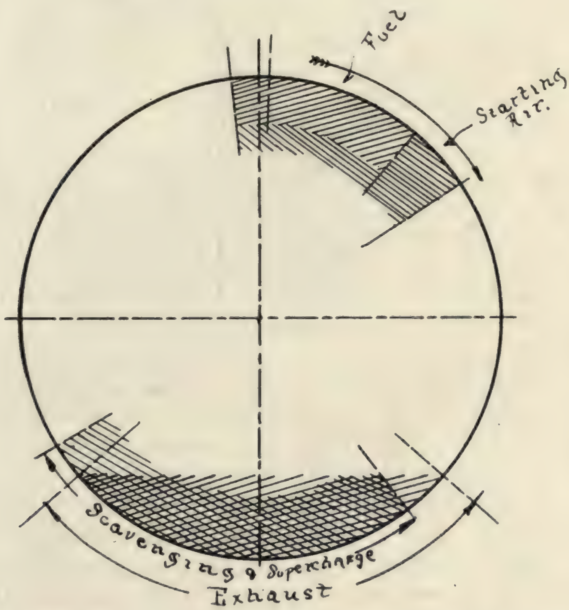
Ignition Failures: There are numerous causes of ignition failures on Diesels. If, when attempting to start the engine, ignition fails to occur, it may be attributed to one or more of the following causes: Low Compression; Cylinders too cold; Insufficient fuel; Fuel injection too late; failure of spray-air supply. If the engine cannot be brought in

motion after repeated attempt, an investigation should be made and causes determined and the same be remedied.

The compression in the cylinders should in first place be given due attention. If the compression is not sufficiently high, the desired temperature necessary to ignite the oil is too low. In many instances this is due to leaky cylinder head or valve cage gaskets. If leaky relief valve, this defect will be noticeable by the noise of escaping air.

Lack of sufficient fuel or total failure of supply to cylinder may be caused by an empty fuel service tank or through stoppage of fuel in the fuel line between measuring pump and tank. Examine all valves.

On some types of fuel-measuring pumps the air-starting gear and pump mechanism are not interlocked in such a way that the pumps are automatically put into operation when the engine begins to turn by air. In this case it may happen that the pump levers are not properly brought in the operating position before starting.



Valve Settings of Valve-Scavenging Two-cycle Engine

Because of the small quantity of oil handled per stroke by the fuel measuring pump and the high pressure pumped against, this pump is very sensitive to air that may be present in oil. A fundamental requirement in good pump design is that no pockets may be permitted in the oil passages in pump or valve chamber, where air might collect but many pumps have been and are still being built that do contain such pockets. Most pumps are provided with vent valves so that the collected

air may be blown out. Sometimes the fuel may be prevented from reaching the pump by an air pocket in the pipe between the pump and the tank.

A very common cause of failure of fuel supply is leakage of air past the check valves. In all closed nozzle-type spray valves the mixing chamber in the valve body, where the spray air and oil mix before entering the cylinder, is always in direct communication with the spray-air system and consequently is filled with air at injection pressure. In order for the fuel pump to force the oil into this chamber against the air pressure, it is essential that the oil pipe be full of oil right up to the inlet to the valve chamber, so that when the pump forces a small amount of oil into the pump end of the pipe, an equal amount will be forced out of the other end into the valve chamber.

It is obvious that if this pipe is partly filled with air, the oil column, when acted upon by the charge of oil being forced into the pipe by the pump, will simply compress the air and no oil will be discharged into the valve. If the discharge valve of the fuel pump is perfectly tight, no air from the spray-valve chamber can force its way into the oil pipe after the pipe is completely filled with oil, but the fine grit present in nearly all fuel oil makes it very difficult to keep this valve perfectly tight very long. For this reason practically all Diesel engine builders install a check valve in the oil line to each spray valve, as close as possible to the point of entry of oil into the spray-valve body. This valve closes against the air pressure in the spray-valve body so that the oil column in the pipe is subjected to pressure only during the time the pump is discharging into the line at one end and forcing the oil through the check-valve at the other. With this arrangement a pump will work quite satisfactorily even though the discharge valve is not perfectly tight, as long as the check-valve remains tight.

The spray-valve should be tested occasionally. This valve can be tested while the engine is stopped, by turning spray air from the bottles into the air line to the spray valve and then opening the by-pass in the fuel-oil line near the check valve. If the valve leaks, the air will blow out of the by-pass. If there is no by-pass in the line, the oil pipe may be disconnected at the pump and the air will blow out there.

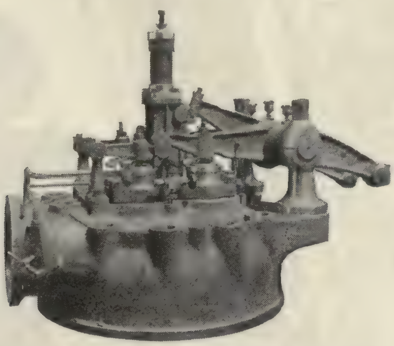
Before making this test, the engine must be jacked around until the spray valve, to which is attached the line being tested, is in the closed position, so that the spray air will not blow into the cylinder. In order to provide additional insurance against spray air leakage into the fuel lines, some builders provide two check valves in each oil line and two discharge valves in each pump.

Leaky suction valves in the fuel pumps, or valves stuck open, may be responsible for the failure of the oil to reach the cylinders. Examination of the valve and seats will usually indicate a leaky condition.

When the fuel contains considerable water, the water may settle to the bottom of the supply tank, while the engine is stopped, in suffi-

cient quantity to fill the pump, so that water instead of oil will be injected into the cylinders. The obvious remedy for this is to drain all the water out of the system before attempting to start the engine.

If the fuel is not injected into the cylinders until after the compressed air has started re-expanding as the pistons move away from the heads and increase the cylinder volume, the temperature of the air may have fallen so low that it will not ignite the oil, and the effect produced is the same as in the case of low compression. This late injection may be caused by the adjustable nose on the spray-valve operating cams slipping. The clearance between cams and rollers may be too great or the valves may be clogged so that the fuel does not flow rapidly enough. The cams should be examined to see if they have slipped on the shaft; if they have not, then the cam toes may need advancing by means of the adjusting screws. The rollers should be examined to see if any are badly worn or broken. Each valve should be checked with the dial plate or the valve-setting marks on the flywheel.

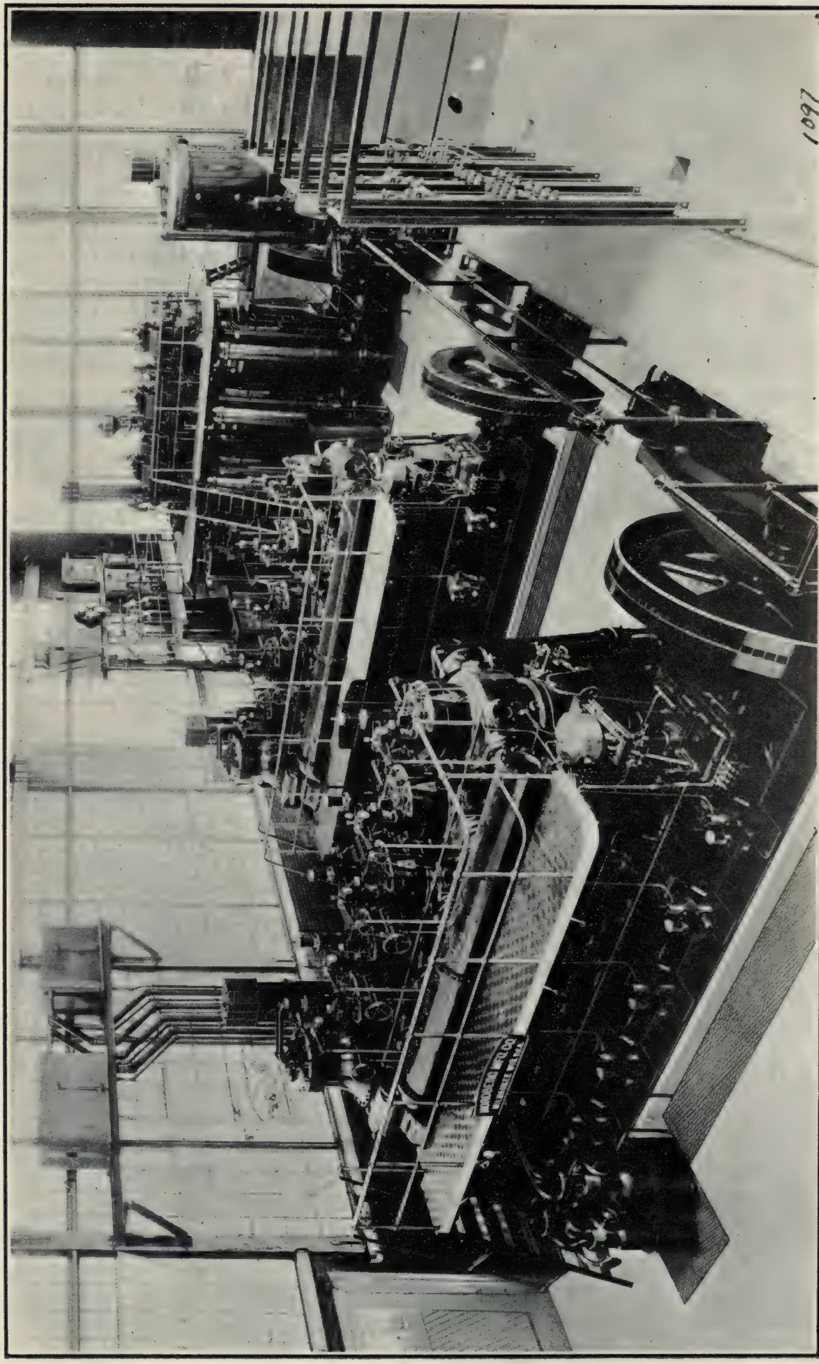


Top view of E. G. Cylinderhead (Nordberg Engine)

If the spray-valves are clogged so that the fuel is retarded in its passage through the valves, an abnormal rise in spray-air pressure will be noted if the compressor suction is open wide when the engine is turning on starting air.

The capacity of the spray-air bottles is often so small that if the spray-air compressor does not begin charging immediately upon starting the engine, the result will either be complete ignition failure or ignition will occur for a few revolutions, then fail as the pressure in the bottles falls. When this occurs, no further attempts to start should be made until the compressor trouble is located and remedied.

The most common cause of loss of compressor capacity is broken or leaky valves. The location of the defective valve may be determined by observing the gage pressures in the different stages while the engine is turning over. An abnormal rise in pressure in the first or second stage indicates that air is leaking back through the discharge valve in



Installation of Nordberg Diesel Engines in Oklahoma. Two in foreground connected to generators. One in background connected to Nordberg Two-stage air-compressor furnishing air for hammers in forge shop

that stage. Rise of pressure in the high stage may indicate a closed stop valve in the discharge line to the engine, clogged strainers or clogged spray valves.

If an excessive amount of lubricating oil is used in the compressor, a jelly-like emulsion will be formed, which will lodge in the strainers and interfere with air flow. If the compressor shows loss of capacity, with pressure below normal in all stages, it may be due to obstruction of the suction of the first stage. In the case of compressors that are regulated by throttling, this suction loss of capacity may be found to be due to the suction valve being closed.

Another cause for rapid loss of spray-air pressure is sticking of spray valves. If a spray-valve stem jams in its guide so that the valve is not forced back to its seat by its spring, the spray air will blow into the cylinder during the whole cycle and so much air will be blown away that the pressure in the system will fail. A condition of this kind will make itself known by very severe explosions in the affected cylinder, due to pre-ignition of fuel that has been blown into the cylinder too early in the cycle.

If the jacket-water circulating pump is started before the engine, it may happen that the cylinder walls and cylinder heads may be chilled to the point where ignition is interfered with, this condition being most likely in cold climates, during the winter months when the cooling-water temperature is very low. This cooling affects the ignition in two ways; it reduces the temperature of the compressed air in the engine cylinders and it also increases the viscosity of the fuel oil after it is deposited in the spray-valve cavity, so that atomization of the oil is more difficult and its passage through the valves is retarded. In cases, where this trouble is experienced, it is best not to start the cooling-water circulating pump until after the engine is started. When steam is available, it is advisable to make a connection to the water system so that the circulating water may be heated and the cylinders warmed up before starting the engine.

FUNCTION OF FUEL INJECTION PUMP

Inasmuch, as the fuel delivered to the Combustion Chamber must be in excess of the pressure (from 45 to 75 atmosphere—i. e., 640 to 1,100 pounds per square inch) in the valve casing, due to the fuel injection air, the pump in itself has to be exceedingly strong and above all mechanically well proportioned.

Properly speaking, the pump regulates the running of the engine, delivering the exact amount of fuel necessary on the combustion stroke corresponding to the load capacity of the engine.

It will be seen, from the detailed description of the different makes of engines explained in this book, that the design of the pump differs but very little.

The piston is always of the plunger type, made of steel; the valve of bronze material, cast iron or steel, with conical seatings, one suction and one, or two in series, for delivery, loaded with light springs, and accessible for immediate examination or where the requirements of cleaning or grinding calls for it. The joints of the copper delivery pipes are usually made with conical connections.

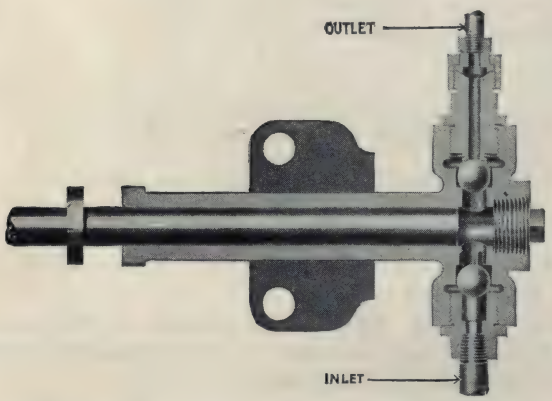
Almost every pump is of a very massive design, manufactuured of cast iron body; the plunger and other moving parts withstanding pressure have carefully packed glands.

The pump, which acts under control of the governor according to the load on the engine requires careful attention. In particular, this is true when the engine runs under low power, lightly, endeavoring to supply the dense and viscous fuels employed. A very small bubble of air in the pump chamber sometimes is the cause of stopping the action of the pump. The plunger in its slow motion merely compresses and expands the pocket of air without causing the valve to raise.

It appears to be difficult to design a pump overcoming reaction of the engine governor caused by variation of the speed requirement of the engine. These difficulties usually are overcome by the method of variation of plunger pump stroke.

While in many cases the regulation is not obtained by an alteration of the plunger stroke, but a quantity of oil corresponding to the whole pump cylinder volume passing the suction valve each suction stroke.

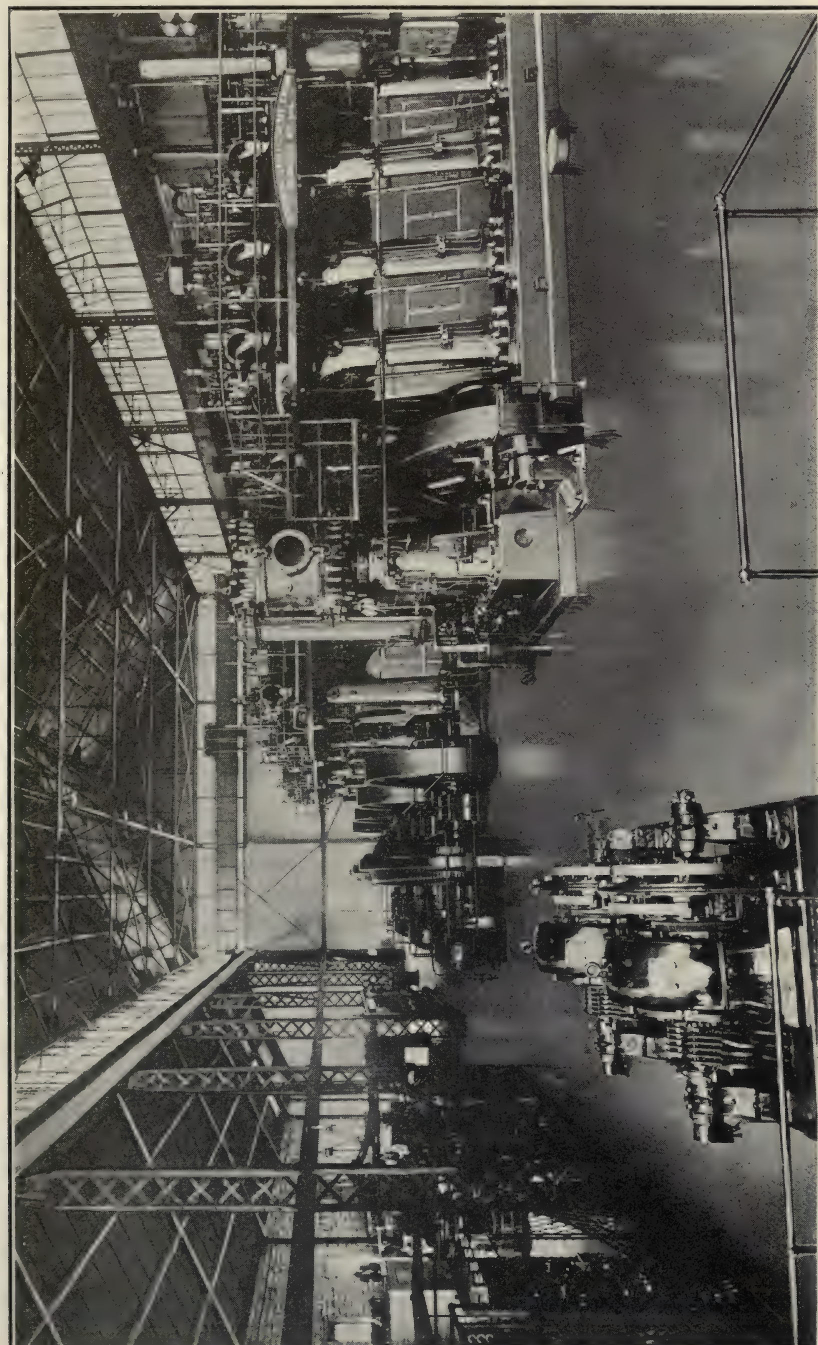
This explains the reason why the fuel injection pumps of Diesel engines draw an excess quantity of oil than actually required, a part of this goes to the fuel injection valve, the surplus passing back through the suction valve during part of the period of the delivery stroke.



Oil Injection Pump of the Giant Oil Engine

An oil pump, as used on the Giant Engine, is shown in the illustration, this pump is operated by means of the eccentric, rocker, cam, and the pump rod.

The quantity of oil injected into the cylinder at each stroke of the piston is determined by the length of the stroke of the pump plunger. The length of



Four Nordberg Diesel Engines direct connected to Generators. In foreground one Engine connected to Nordberg air Compressor. This Installation is in the Southwestern Copper District.

this stroke is, in turn, determined by the position of the plunger cam, which in turn, is determined by the speed of the engine governor.

Before attempting to start the engine, the pump should be thoroughly cleaned. This is best done by unscrewing the plugs at the top and bottom of the pump body, removing the steel ball valves, and washing out thoroughly with gasoline or kerosene.

In most engines a special reservoir is provided, which usually is first filled by means of a hand lever. Before starting, make sure that all the oil pipes and connections are clean, and then assure the tightening of all joints eliminating all possibilities of assembling of inside-air, which as previously explained, may cause air pockets.

The importance of perfect tight joints, owing to high compression pressure must be emphasized. Leaky valves or connections are frequent occurrences, in particular where they are in contact with high pressure.

Satisfactory operation of pumps and all mechanical contrivances depends in most every case on the operator and the safest method of assurance in proper operation of the plant is to be alert at all times. The cleaning of valves is a necessary matter which should never be neglected, in particular where fuel oils are used with ash ingredients.

FUEL PUMP AND CONTROL END OF WORTHINGTON 2-CYCLE DIESEL ENGINE

Referring to illustration, showing outline cut of fuel pump and control end of four-cylinder engine, speed regulation is obtained by opening a by-pass and not by variation of the length of the fuel pump stroke.

The amount of fuel supplied to the cylinder depends on the time of opening of the by-pass valve. This in turn depends on the angular position of the eccentric shaft, which is controlled by the governor.

The governor, which is located on the end of the engine crank shaft, is connected to the eccentric shaft by suitable links. Any increase in the engine speed from normal will cause the governor to turn eccentric shaft through a small angle which at the same time will lift end of by-pass lever. When the fuel pump plunger raises the by-pass lever and by-pass plunger, by-pass valve will be opened earlier. As a result of this earlier opening of the by-pass valve, more fuel is by-passed back to the fuel supply reservoir, thus reducing the amount supplied to the cylinder and promptly bringing the speed back to normal, withont changing the time when injection starts.

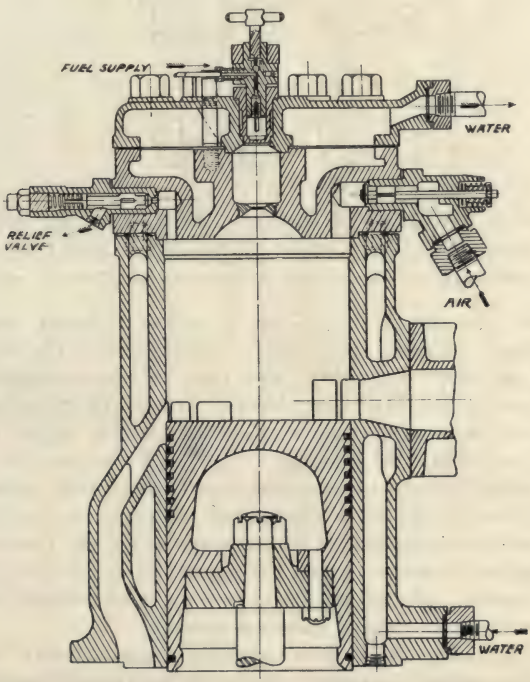
Eccentrics keyed on the engine crankshaft drive the fuel pump plungers through tappets, as shown. The upper ends of the eccentric straps are provided with hardened steel contact rollers and are guided by links, replacing the crosshead and guide construction previously used. The pump plunger tappets pass through a partition, which prevents fuel

oil leaking down into the control housing and mixing with the lubricating oil. All running parts are splash lubricated by oil from end main bearing, overflowing back to the crank case pump so as to keep a high level in the control housing.

A hand adjusting screw at the end of the by-pass lever makes it easy to equalize the oil delivery from all plungers on multi-cylinder engines. Pump plungers take oil from a constantly full suction tank with a strainer, through which fuel oil is circulated by the fuel oil supply pump.

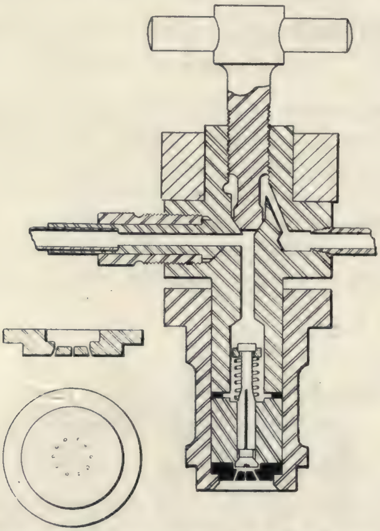
COMBUSTION CHAMBER AND SPRAY VALVE, WORTHINGTON 2-CYCLE DIESEL ENGINE

Referring to illustration of cylinder head and spray valve, the operation of the spray valve is extremely simple. The check valve back of the spray orifice disc is held on its seat by a light spring and lifted very slightly by the oil flow pressure at each delivery stroke of the pump. The oil is distributed to ten small holes arranged in a circle and one at the center. These produce ten slightly diverging high velocity jets that break into spray near the injection orifice merging one into the



Section of Cylinder and Head of Worthington Diesel Engine Two-cycle Solid Injection)

other and with the center jet. The amount of fuel injected is controlled at the fuel pump by a by-pass valve which opens at a variable point of the stroke to stop delivery of oil. The oil pump is operated by an eccentric on the end of the crank shaft. Oil delivery always starts at the same time, i. e., when the fuel pump tappet strikes the pump plunger. This occurs at a time when the motion is rapid, so as to secure a quick, sharp injection. Fuel in a finely divided state, is sprayed directly into the injection chamber when the compression is high enough for the air to ignite the fuel. This chamber is completely water jacketed.



Exposed view of spraying arrangement as used on Worthington latest Two-cycle Solid Injection Engines. It should be noted here that engines of the Worthington type can be manufactured from one H. P. up.

The small amount of air in the injection chamber receiving the full fuel charge, permits only part of it to burn, gasifying the rest, and without any shock pressures. The form of fuel oil spray is such as to use only part of the injection chamber air during injection. The unburned fuel and unused air pass through an ejection orifice to the combustion chamber when the pressure in the injection chamber is greater than in the cylinder, and complete burning takes place during the first part of the downward stroke of the piston. The rate of circulation in the cylinder is mainly controlled by the movement of the piston itself.

The compression pressure, and the maximum combustion pressure do not normally exceed 450 and 500 lbs. per square inch, respectively and the latter may even be no higher than the former.

The non-explosive combustion, without any possibility of explosive shock pressure, makes the expansion as smooth as possible under any condition. The engine being two-cycle, every outstroke is the same, and this combined with the compression on every instroke adds greatly to the perfect operation of this engine.

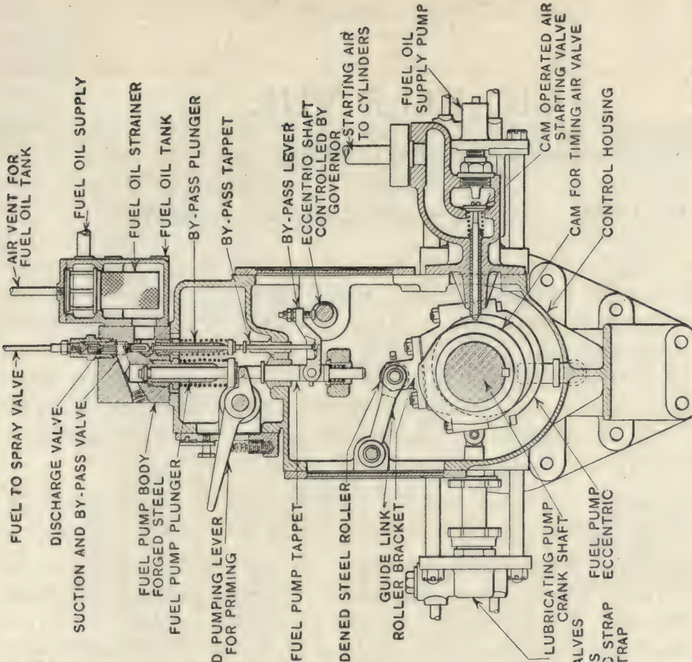


Fig. B – 3130

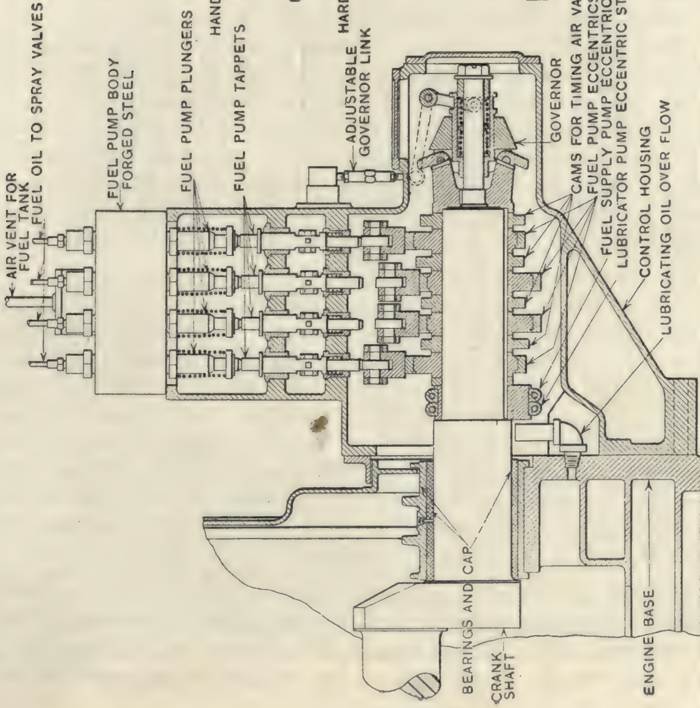


Fig. B – 3131

Worthington Diesel Engine. Two-Cycle Soil Control End of Four-Cylinder Engine, With Details of

CHAPTER VIII.

PRINCIPLES OF CONSTRUCTION

TWO-CYCLE vs. FOUR-CYCLE DIESEL ENGINES.

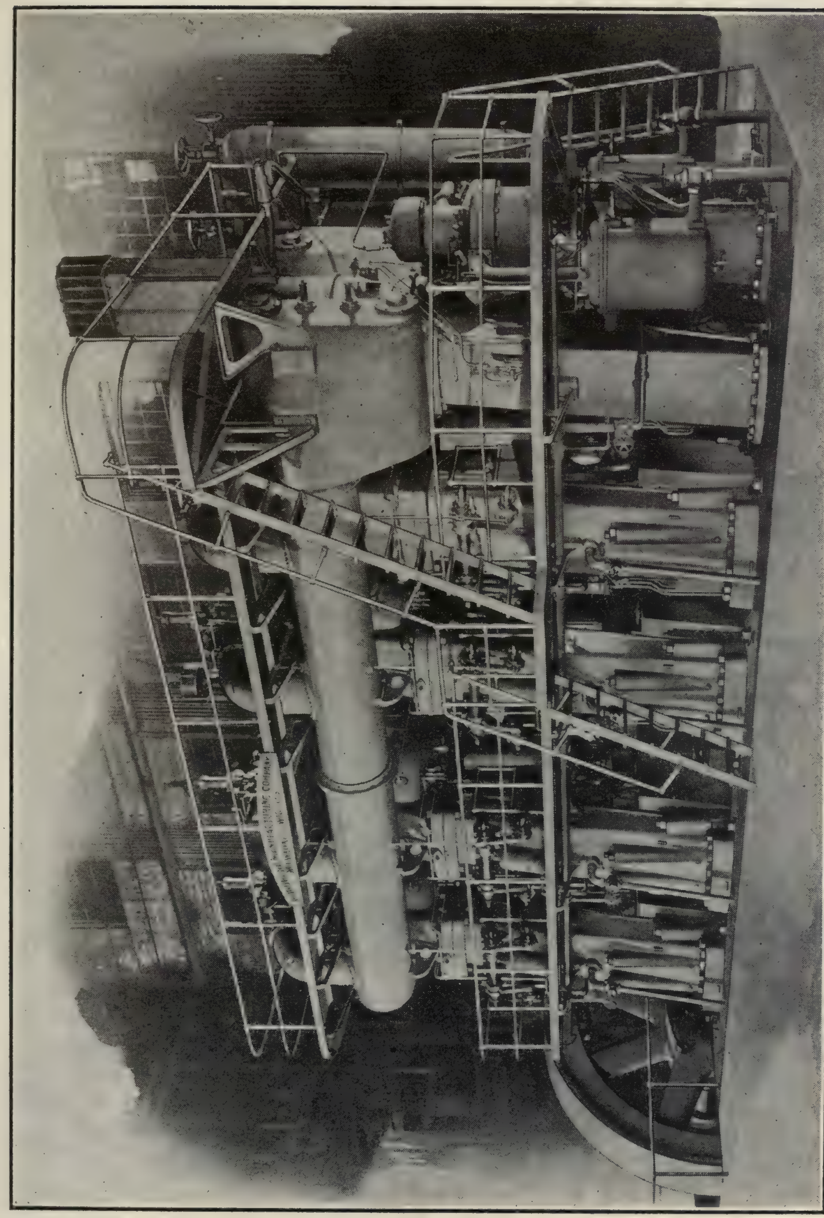
The relative superority of two or four-cycle internal combustion engines for marine purposes is one of the most debated questions at the present moment from a theoretical as well as from a practical standpoint; thus it forms daily the subject of discussion, lectures and articles in technical review. The chief purpose of this article is to co-ordinate the arguments which have been alleged for and against both types in their best form of construction, and to endeavor to draw a conclusion after careful consideration of all points of the question.

The advantages which are usually attributed to the two-cycle engine as compared with the four-cycle type may be briefly stated as follows:

(A)—The two-cycle engine developes a greater power than the four-cycle with the same number and size of cylinders and the same number of revolutions. This advantage of the two-cycle types is due to the fact that the four-cycle type gives an impulse for each cylinder every two revolutions, while the two-cycle type gives an impulse each revolution, theoretically the two-cycle type should therefore develope, under the same conditions, a power double that of the four-cycle type. In practice, however, the said theoretical limit has never been reached, but at present it may be said that the power developed by a two-cycle engine is 175 per cent, to 190 per cent of that of the four-cycle engine, and it may be added that while the mean effective pressure in the four-cycle type is about 5 kg. per cm. ^{2}(71 lb. per sq. in.) that of the two-cycle is practically of 4.4 kg. to 4.75 kg. per cm. ^{2}(62 to 67 lb. per sq. in.).

The essential advantage of the two-cycle type brings as a consequence a remarkable reduction of space and weight, which may be approximately calculated in the following manner: As there is no reason that a four-cycle cylinder with its framing and driving gear (assuming the same intensity of stress of the materials) should weigh less than a two-cycle cylinder of the same size, and as the weight can be practically considered to be proportional to the volume swept by the piston, therefore, for the same power and number of revolutions, the cylinder of the two-cycle engine (175 per cent being taken as the power ratio of the two-cycle to the four-cycle type) has a weight which is 57 per cent of that of the four-cycle engine.

This average is somewhat reduced by the fact that the two-cycle engine needs scavenging pumps. and as, according to circumstances and to the different design of the pumps, their weight can be considered as being 8 per cent to 12 per cent of the weight of the cylinders, it



2000 B. H. P. Nordberg Two-cycle Diesel Engine. Longest in America. With the suprisingly low fuel consumption and operating for months with little attention this type is well deserving of admiration.

results that the weight of the two-cycle type will be 62 per cent of 65 per cent as compared with the weight of the four-cycle. The above figures seem also practically confirmed, though there is always some difficulty in comparing numbers quoted by different constructors, for they do not always state which parts of the equipment of the plant are included or excluded from the figures published. But besides the saving of weight there is also the saving of space.

It must be noted that the saving of space by the two-cycle types has also as a consequence a considerable saving in the cost and weight of the engine seat as well as in the dimensions of the engine-room, facilitating the supervision and control of the machines.

(B)—The turning-moment in the two-cycle engine is far more regular (for the same number of cylinders) than in the four-cycle type; the results of even the four cylinder two-cycle type are far more regular than those of the six-cylinder four-cycle engine.

This advantage of the two-cycle engine is not merely theoretical, but in practice results in a minor intensity of the vibrations of the stern end of the ship, besides a reduction in size and weight of the line of shafting and consequently of its fittings, such as supports, stern tube, etc. According to Lloyds Register the section of the shafting of a six-cylinder four-cycle engine (for the same power and the same number of revolutions) ought to be 45 per cent, greater than that of the six-cylinder two-cycle engine. Furthermore, the reduced size of the flywheel in the two-cycle engine and the reduced space permits of placing the engine nearer the stern, not only saving in the length of the line of shafting, but also increasing the space available on board for the cargo.

(C)—The two-cycle engine offers greater facility in reversing as compared to the four-cycle type, which is due to the fact that in the former the exhaust of the burnt gases takes place thru ports in the cylinder wall, so that in order to reverse the running of the starting valves, the alternation in the timing of the scavenging valves is very readily made by rotating the cam-shaft relatively to the crankshaft, while the alteration in the timing of the fuel and starting valves (these valves having but a small lift) can be readily effected by employing double cams sliding on the shaft.

In the four-cycle type on the contrary, besides the alteration in the timing of the fuel and starting valves, it is necessary separately to reverse the inlet and exhaust valves; and as the latter operation requires a different rotation on the cam-shaft, it is not possible to employ the simple device of the two-cycle type, but much more complicated mechanism becomes necessary.

Referring further, to the starting and reversing devices, it may be added that the necessity of being able to start the engine whatever be the position in which the cam-shaft has stopped, that phase of the starting air does not permit of a reduction in the number of cylinders

to less than six in the four-cycle type, while the two-cycle came can be constructed with but four, and be kept in its perfect manoeuverability.

(D)—With the two-cycle engine the inertial of the reciprocating parts such as connecting rods, pistons, etc., is balanced at top-dead center by the pressure on the piston, which cannot be realized in the four-cycle for the exhaust and suction strokes; as a consequence, in the four-cycle type in order to avoid the possibility of their breaking and the great damage this would cause.

(E)—The two-cycle engine does not require any exhaust valve for the burnt gases, and in the engine provided with port scavenging there is no need of any valve subjected to the action of the burning gases; in the four-cycle type the exhaust valves are the source of well known troubles and even in the case their tightness and durability is increased by using more or less complicated cooling devices, the danger of their falling into the cylinder, with all its serious consequences, can never be fully eliminate.

It should be noted that the exhaust valves of the four-cycle engine are the parts which are the most sensitive to the quality of fuel and are especially liable to suffer by the asphaltum and sulphur sometimes present in heavy oils of certain origins. For a two-cycle engine without exhaust valves there may consequently be used certain kinds of fuel which are not suitable for a four-cycle engine.

Against the advantages above referred to as to the two-cycle type, the advocates of the four-cycle engine oppose some objections which partially apply to all two-cycle engines; and partially apply to special types or to constructive details of them. These objections may be briefly stated as follows:

(a)—In favor of the four-cycle type it has been said that the experience of the gas engine has lead back again (after a period of preference for the two-cycle engine, so that it is convenient to select again the four-cycle type).

Against this objection we may note that the example of the gas engine, as compared with the four-cycle, shows the disadvantage of a greater consumption and of the inefficient regulation at light loads; the greater consumption being due to the fact that a certain amount of gas is always mixed with the scavenging air because the two fluids cannot remain wholly separated, and so unburnt gas escapes with the air thru the exhaust ports without producing any useful work. The bad regulation is due to the difficulty of having the right mixture in case of light loads, because in the two-cycle engine it is impossible to regulate the power without diluting the explosive mixture. Neither of the said inconveniences exist in the Diesel engines, the scavenging being made with pure air and the regulation being obtained in exactly the same manner in both the two-cycle and in four-cycle types. Moreover, it may be stated that notwithstanding the said inconveniences, which cannot be neglected, the gas two-cycle engines are still constructed, and

in work for many hundred-thousands of horse-power, from which we may draw the conclusion that the two-cycle engines offer other real advantages.

More suitable than the example of the gas engine for comparison is that of the hot-bulb engines where the two-cycle type is preeminent, for the Bolinder, Skandia, Fairbanks-Morse, Petter, Torbinia types, a. s. o., have almost completely eliminated the competition of the four-cycle type, especially for high power.

Referring now to some failures of the two-cycle Diesel engine, it may be said they are mainly due to constructive defects; numerous inconveniences have been experienced in the engine with stepped pistons, and it would therefore be wrong to attribute these failures to the type of the engine in itself, instead of to defects in design.

The supporters of the four-cycle type allege that the two-cycle engines are far more complicated, not only on account of the scavenging pumps, the piping and the receivers relating thereto, but also on account of the greater complexity of the valve gear.

Against this assertion it may be objected that the air pumps which undoubtedly constitute an added organ, by no means interfere with the reliability of the working of the engine, as they are always working at very low pressures and temperatures, like the low-pressure cylinders of steam engines; and constructively it is certainly more rational to employ a suitable air pump instead of using, for half the time, for displacing the air, enormous pistons which have been designed and fitted with rings for at least a hundred times higher pressure.

Referring now to the valve gear, the complexity pertains exclusively to that two-cycle type of engine having scavenging valves in the cylinder heads, whilst in the recent type with port scavenging, besides the fuel and the starting valve (like that of the four-cycle type), there is only the scavenging valve to control. This is light and easily displaced, as it is not subjected to the highest pressures and temperatures of the cycle, and it does not require to be perfectly tight. This valve can easily be replaced by a rotary valve. In the cylinder of the four-cycle engine, instead of one scavenging valve there are two at least to be controlled, and very often two inlet and two exhaust valves, which, being placed in the combustion chamber, require to be perfectly tight and need an precise and reliable operating gear in order to withstand the effort of the powerful closing springs.

In favor of the four-cycle type it has been furthermore affirmed that its fuel consumption is far lower than that of the two-cycle engine. Now even, if it must be admitted, that this objection is correct in relation to the first two-cycle engines which were constructed, and is also applicable to some present motors of defective construction, it has, nevertheless, lost much of its importance when comparing the four-cycle engine with the best known modern two-cycle engines.

It is true, that some excessively low figures have been singly re-

ported for the consumption of four-cycle engines, but they can be safely overlooked upon consideration of the circumstances of the test or of the uncommonly high consumption of the lubricating oil, which has obviously partially burnt as fuel, so that the above stated results can be quoted as corresponding to the best up-to-date constructions. Though they still show a slight advantage for the four-cycle engine, this is no greater than 3 per cent, or 5 per cent, and if we consider the other element required for calculating the real working expenses, this difference is not of great importance. It must, indeed, be noted that the installation of two-cycle instead of four-cycle engines for a given type of ship, results in a saving in weight and space, and therefore a reduction of displacement and the possibility of increasing the run of the stern (this leading to a reduction in the power for propelling the weight and the space taken by the propelling plant) have the greatest influence. Furthermore, it may be added that, even if the question of the weight and space should be regarded as a secondary one, still the two-cycle engines show the advantage that the particulars being the same, it can develope the same power as the four-cycle one at a much lower speed revolution, with the consequence of rational and systematic experiments, in a few years, from 250 grams or 260 grams per brake horse power, to the present values, it will still improve until it reaches and even surpasses the low consumption of the four-cycle type. Theoretically, there is no reason why this should not happen, for the thermal efficiency is the same in both types, and the power required by the two-cycle engine cannot be greater than the power expended in driving the main pistons of the four-cycle engine to work half the time as pumps themselves.

Finally, besides the fuel consumption, that of the lubricating oil, which is much more expensive, ought to be considered. It is obvious that the two-cycle engine should require a less quantity of oil than the four-cycle, the load on the piston of the four-cycle engine being 50 per cent greater (with the same number of cylinders and the same ratio between diameter and stroke) than that of the two-cycle, the pressure exerted on the bearings, and on the guides being proportionately increased so that the surafec to be lubricated is accordingly larger. In practice, however, as the two-cycle engine may be constructed with fewer cylinders the saving in the lubricating oil is still more evident. At present the figure of 3 grammes to 4 grammes (0.00614 lb. to 0.008818 lb.) per brake horse-power as the total amount of oil consumption is usually reached in high speed engines (480 revolutions).

As another advantage of the four-cycle type, it is affirmed that the cylinder wall never reaches such high temperature as in the two-cycle type, so that the latter are subjected to higher internal strains and thus to the danger of cracks. Now, while it is true that the ratio between the quantity of fuel burnt in the four-cycle type and the surface of the combustion chamber is hardly superior to one-half the same ratio in the two-cycle engine, other important circumstances have been overlooked which have certainly a great influence on the mean temperatures.

The action of the hot gases on the cylinder walls lasts certainly a shorter time in the two-cycle than in the four-cycle type. While in the latter the cylinder walls undergo the action of the hot gases during the whole expansion and exhaust strokes, that is, practically for more than half the time, in the two-cycle engine the action of the hot gases lasts only for a little more than two-thirds of the working stroke,

In the two-cycle engines in which the exhaust occurs thru ports, the latter open much more rapidly than the exhaust valves of the four-cycle engines, and consequently there is a much more rapid diminution in the temperature due to expansion.

While the exhaust temperature in the four-cycle engines is seldom below 350 degrees D. and in the high speed engines is easily reached 450 deg. or 500 deg. C., in two-cycle engines, if well constructed, this temperature usually remains under 250 deg. C., and sometimes it only reaches 200 deg. or 210 deg. C.

Not one of the hypotheses above referred to is in the favor of the two-cycle engine; the hypothesis of the same initial compression temperature in both types is unfavorable for the two-cycle type, as all experiments which have been made with gas engines confirm that in the two-cycle engines a much higher compression ratio can be employed than in the four-cycle engine, without the danger of pre-ignition, and that the mixture in the beginning of the compression is therefore cooler in the two-cycle type. By measuring the diagrams with a plainmeter, however, the conclusion was reached that the mean temperature of the two-cycle is practically the same.

Taking account of all these elements it is fair to say the two-cycle engine, from the standpoint of temperature, is in better condition than the four-cycle. The two-cycle engine, in which the inner walls of the cylinder, after the very short action of the flame, are immediately colled by the scavenging air current (which is supplied in such quantity as to allow, besides the filling up of the cylinder, the escape of the warmest portion which entered at first) is thermally superior to the four-cycle engine, in which all heat must be abstracted thru the walls of the cylinders with the consequent fall of the temperature in the walls and resultant internal stresses.

The opponents of the two-cycle engine allege that the engines of this type some portion of the combustion gases remains in the cylinders, especially in the upper part of them, so that the cylinder head becomes excessively hot. Against this argument it must be first remarked that in the foor-cycle engine at least 8 per cent of the burnt gases remain to fill the compression chamber when the piston has completed the exhaust stroke, and it is obvious that this remaining portion cannot but contaminate the air which is drawn in during the subsequent stroke. As regards to the two-cycle engine the assertion that some residue of the burnt gases still remain in the cylinder after the scavenging operation is merely a gratuitous hypothesis, which is contradicted by the facts above referred to, according to which the quantity of heat absorbed by

the walls is less than in the two-cycle engine, and that in the two-cycle type the compression ratio can assume a greater value in the four-cycle engines.

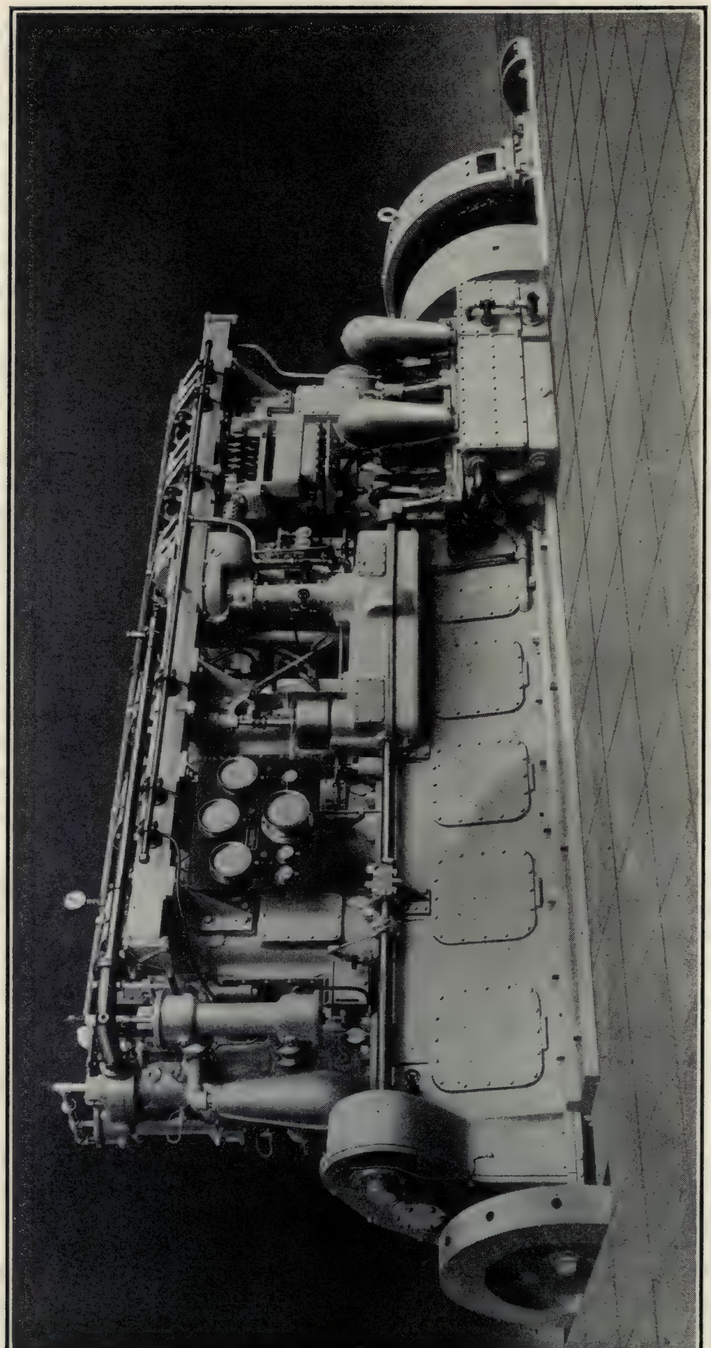
Against the four-cycle engine it has been said that the four-cycle type can run with greater regularity than the two-cycle when working at low speed of revolutions, owing to the fact that in the two-cycle engine the compression at low speed falls rapidly with the diminishing of the scavenging air pressure. It must, however, be noted that this observation is correct merely when it refers to two-cycle engines of bad design, in which, owing to inefficient construction, the scavenging air pressure rises, at the normal speed, to excessively high value, while in the two-cycle engines, which have been carefully designed even at full speed the pressure of the scavenging air remains within very small limits. By the speed reduction the pressure is also somewhat reduced, but not so as to cause failure of the ignition especially when the engine is hot. Practically, in both the two and four-cycle types, the lowest limit of speed is dependent upon the two-cycle engines is more than efficient for perfect manoeuvering. Moreover, it must be remarked that the turning moment of two-cycle engines being more regular, and it being possible to run with half the number of cylinders and to obtain sufficiently good regularity, the two-cycle engine shows in this particular point an advantage compared with the four-cycle type.

Authors Note: In above article it should be noted that comparison of the two-stroke-cycle vs. four-stroke-cycle type of Diesel engines depends a great deal on the view of manufacturers. Each builder naturally stands for the type of his particular make of engine. Both types, as will be seen, have their advantages and also disadvantages, depending on the class of work they are performing. While the viewpoint expressed in this article represents the stand Mr. Giovanni Chiesa of the Ansaldo San Giorgio Works of Turin, Italy, the stand taken by Mr. Franco Tosi of Legnano, Italy, again entirely claims the superiority of the four-cycle construction for Diesel Machinery, as will be seen in the article dealing with the advantage of the four-cycle over the two-cycle type.

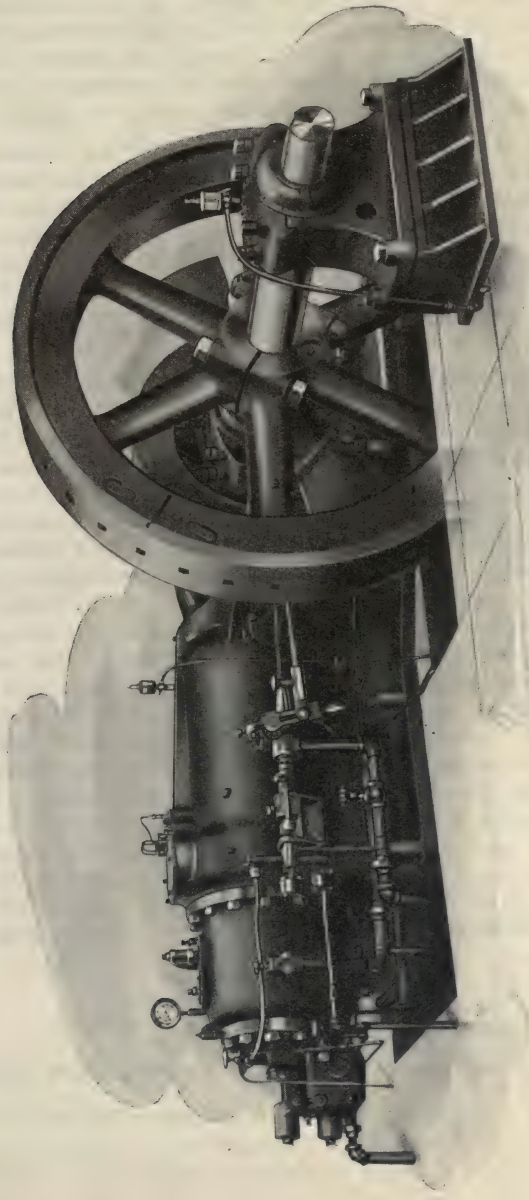
POINTS OF ADVANTAGE AND DISADVANTAGE OF TWO-CYCLE IN COMPARISON TO THE FOUR-CYCLE DIESEL ENGINE.

In this section dealing with the advantages claimed on engines built on four-cycle principle, some conclusion may be gained when comparing the arguments advanced by adherents to the two-stroke cycle as set forth in previous pages.

The controversy as brought before the readers of this book, shoulo bring out many points in favor of either engine. For instance, it is claimed by those preferring the four-stroke type, that if the same life is to be obtained from the two-stroke cycle engine in comparison to its rival,



Power and beauty combined. Winton Marine Diesel Engine, Model 40. Eight 12 15/16" by 18" cylinders



A typical "Standard" engine, manufactured by the Hadfield-Penfield Steel Co., Bucyrus, Ohio

any advantage of reduced space and weight which it may possess disappears. Advocates of the two-stroke cycle engine give, as one of the principal reasons in favor of this type of engine, the greater power that can be developed in a given size of cylinder; but this advantage is only obtainable at the expense of the greatly increased temperature of the cylinder, piston heads, etc., which arises from the combustion of the larger quantity of fuel necessary for the increased power per cylinder.

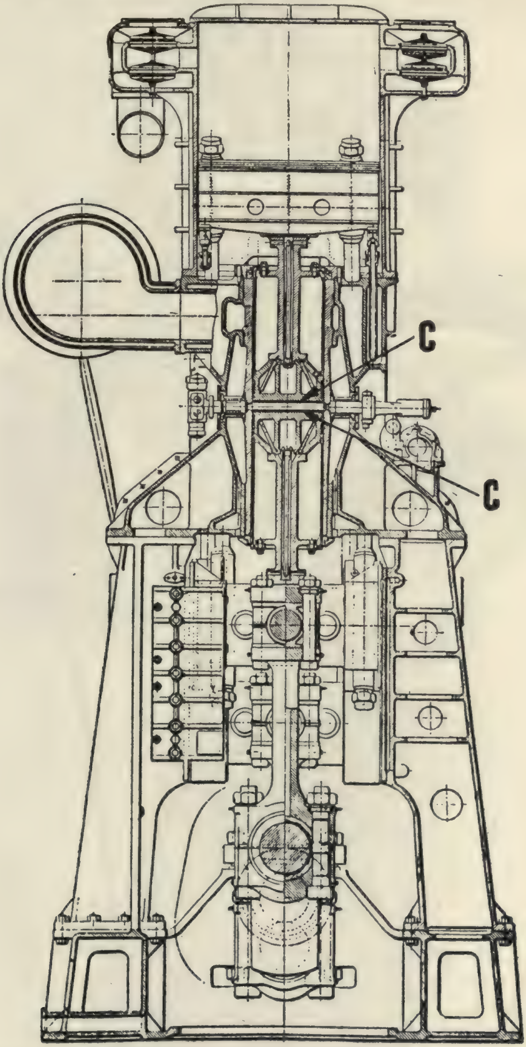
The ratio of increase in consumption of fuel per cylinder of equal size, is, in fact, greater than the ratio of increase of the power obtained from the cylinder, the consumption of fuel per horse-power hour being somewhat greater in the two-cycle engine than in the four-cycle. This greater quantity of heat developed in each cylinder in a given time and the resulting higher temperature is confirmed by the color of the surfaces of the part exposed to it, and is responsible for the trouble that have been experienced in the two-stroke engine.

It follows, therefore, that the two-stroke cycle engine, if designed with the same size of cylinder, will have a shorter life and may require more frequent overhauling than an engine of the four-cycle type. This is a fact of great importance not only with heavy oil engines for cargo boat propulsion, which must run uninterruptedly at long periods at full load, but also for the lighter type of engine used for submarines which, although not required to run fully loaded for such long periods, are subjected to high stresses. Considerable progress has doubtless been made with the two-cycle engine, and many improvements introduced into the design, material, cooling of the cylinder and piston, to overcome the difficulties experienced. The modern two-cycle engine is thus undoubtedly more reliable than the older types, but such engines have been in service for too short a time for a definite judgment on them to be arrived at. On the other hand, rapid progress has also been made in the design of the four-stroke cycle engine, whilst many years of highly satisfactory service at full load have already been recorded. Regarded in another way, if an equal life is required from both types of engine—which is called for in both land and marine service—an equal quantity of fuel should in both types be consumed in a cylinder of a given size, in order to secure equal temperatures. If this condition is adhered to the result will be that the same power will be developed by both types with an approximately equal weight and space.

PRINCIPLES OF DOUBLE-ACTING PISTON DIESEL ENGINES

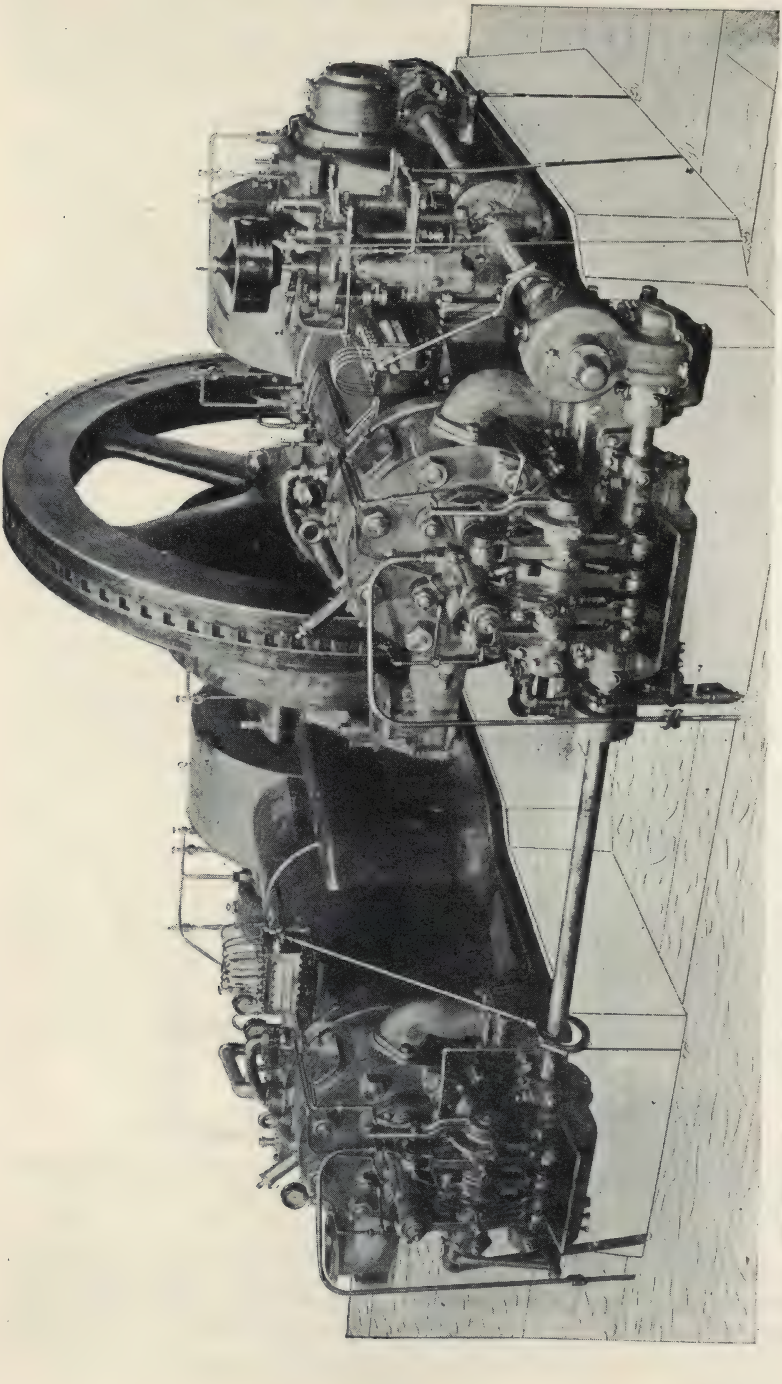
While the single-acting piston Diesel engine has been universally adopted, principally on account of its superiority in regards to simplicity, nevertheless there are factors of numerous advantages in favor of the double-acting piston type. From the earlier experiments carried out in Germany by von Oechelhauser and Junkers, up to the present day, this type has been brought to a highly commendable stage of perfection.

The constructive principles on double-acting Diesel engines prove that there are features which cause a high attaining of power with considerable less fuel expenditure than on engines of single-acting types. On this



The wellknown Junkers engine. A German product which has many advantages as a double-acting-piston engine over her rival the single-acting-piston

account alone an important point is gained, balancing any questionable performance in contrast to the single-acting engine. While experiments with gas-engines, working on the opposed-piston principle, have proven



National Transit Engine of Twin-Engine design. An excellent stationary Diesel engine

highly satisfactory, it should be considered that problems on Diesel engine operation, or constant pressure application, require problems to be solved inherent to this class of engine.

In following explanation it will be observed that there are factors of advantage in favor of the opposed type of Diesel engine.

(1) The subdivision of the aggregate stroke by using two pistons achieves a large ratio of stroke to diameter of cylinder simultaneous with a high rate of revolution, and thus induces favorable conditions in regard to general economic and—due to the favorable combustion chamber—heat economic circumstances.

(2) The use of two pistons and the exhibited solution of the mechanism, connecting them with the crankshaft, enable far reaching balance of reciprocating parts; a far reaching relief of the main bearings; taking up the forces exerted by the pistons in the mechanism itself.

(3) The engine, constructed on the two-cycle principle with its economic advantages, provides good scavenging by the aid of pistons. These, with their large diameter and stroke, represent absolutely ideal governing elements as well as exhaust and the admittance of scavenging air.

(4) The main parts of the engine only experience undue strains and stresses as are created by useful external forces, and are free from uncontrollable and, therefore, causing reliability of operation, heat and internal-strains due to casting.

(5) Higher thermal efficiency with its consequential results of a high utilization of the fuel. Uniform maintainance of heat temperature adding towards regularity. Larger cylinder volume per power-unit convertible into useful work.

The method of tandem-arrangement in opposed-piston practice, has also proven highly satisfactory. In this particular system the two outer pistons act directly, and by medium of the power-transmitting mechanism on the center crank. The two inner pistons are connected by transverse-piece and the side-rods on the side-cranks. These are set 180° with reference to the middle crank. As the particular type under discussion works on the two-cycle, the general arrangement produces double action, i. e., every stroke is a working stroke. While the pair of pistons in the one cylinder are executing an outward movement, i. e., a power stroke, the pair of pistons in the other cylinder approach each other, i. e., to accomplish a compression stroke, and vice versa. When the pistons of the one cylinder have reached their inner dead center, the pistons of the other cylinder have attained their outer dead-center position.

The scavenging process is accomplished in following manner:

At first the one row of ports is opened by the one piston and the spent gases take their exit, seeking an equalization of pressure with that of the atmosphere. Hereupon the other ring of slots is laid bare by the second piston and a quantity of air delivered by the scavenging pumps at low pressure, is admitted to the cylinder. This expels the products of com-

bustion, driving them away in front of itself, leaving the cylinder as free from residue as possible. Thus in this scavenging process the function of governing the ports in the circumference of the cylinder devolves on the pistons. The working-process during one revolution is identical to all two-cycle engines, i. e., (a) admission, (b) compression, (c) power-stroke, and (d) exhaust.

Every cylinder is provided with a fuel-injection valve. The forward cylinder receives in addition a compressed-air starting valve. At both sides of the rear cylinder the double-acting scavenging-pumps are arranged. These are actuated by the middle-traverse piece. In the same line to the rear each of two-stages of the four-stage or three-stage air-compressors are fitted. The valve-gear of the engine is actuated by cams. The driving-shaft runs through under the engine. For each cylinder it drives two short cam shafts carried by rocking frames, for ahead and reversing, respectively. By rotating the rocking supports about the center line of the driving-shaft from the controlling-platform the ahead and reversing cams may be brought into contact with the valve-lever roller, respectively. On each of the cam shafts a starting-cam is situated between two injection cams. The lower fuel-valve on each cylinder is actuated directly by a lever. The other valve is worked by a circuit of rods leading to the top of the cylinder. The moving of the starting valve is effected by medium of an interposed lever. The three rollers for each cylinder are placed axially relatively to one another. The interposed lever for actuating the starting-valve is mounted on an eccentric journal, so that it can be put out of action by the levers leading to the controlling-platform. For reversing it is only necessary to swing the rocking-frames over and keep the starting-valves in action till the engine is rotating in the contrary sense.

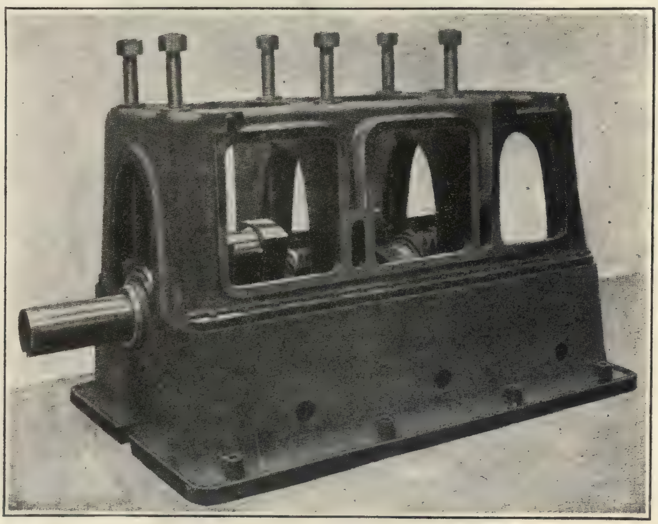
EFFECTS OF INTERNAL AND EXTERNAL STRESSES

The importance of designing Diesel engines eliminating all possibility of undue stresses of either internal or external forces is imperative. Notwithstanding the possibility of restricting the stresses set up in materials of other prime movers to external forces, when the cylinder-walls and corresponding parts are suitably proportioned; it is utterly impossible to obviate stresses produced by internal forces, where there is a permanent heat-flux and a consequent temperature-difference to be dealt with. Such is the case with the cylinder-walls in the internal combustion engine.

The large difference in the heat-transfer per unit of surface at the single sections of the shell must be considered. Under the influence of the temperature-difference the hot cylinder sections tend to expand more than the colder ones. Thus, the shell experiences a strain due to tensile or compressive-stresses. This is again balanced by expansion due to an equalization of heating, an augmented flow of heat is natuarlly to be anticipated. This is to be accounted for by the piston conducting the heat

taken up at its bottom to the external and internal cylinder parts, distributing the heat from the hotter cylinder-sections to the cooler ones, laying more remote from the combustion space.

In a case where the shell contains a hole, the fibre in the vicinity ot the boundary of the perforation experience a stronger strain, causing a consequential danger of cracking of material. The reducing of heat-stresses, inevitable in the internal combustion engine, and, increasing tremendously with the sizes of modern engines of large capacities, to tolerable values, by appropriate measures, is a problem which requires consideration. Combustion chambers should be provided with plain walls, possible to withstand the severest heat-impulses.



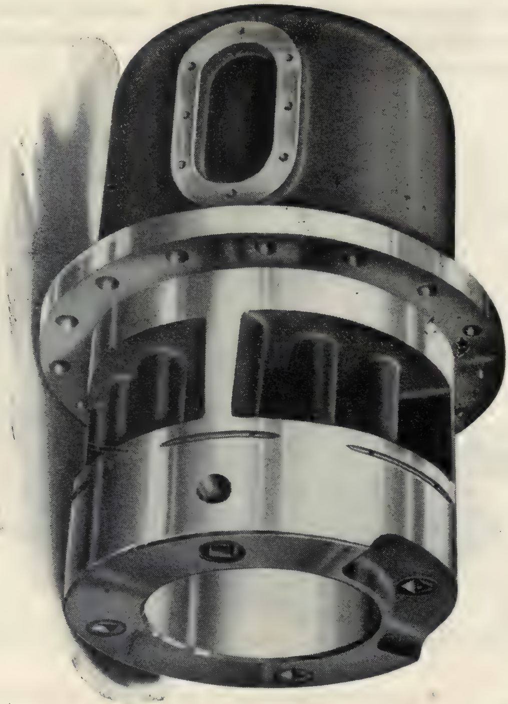
Engine Frame of Standard Engine (Vertical Type)

Correct designs of cooling-arrangements obviating as far as possible existing high-temperatures difference is also a factor which deserves mention. Provisions in designing two-cycle engines, creating a governing element for scavenging, which with great simplicity and reliability of operation would perfectly satisfy the important conditions to be enforced, are of material importance. These consist in opening and closing exhaust and scavenging-areas as rapidly as possible.

It has been found that the design of the cylinder, to correspond with thermal considerations, is, to make them of small diameter and long stroke, so that the heat-efflux during compression, especially in the last part of the same, is curbed as far as possible, insuring positive ignition even at low rates of revolution.

The increased volume-pressure attending the combustion and expan-

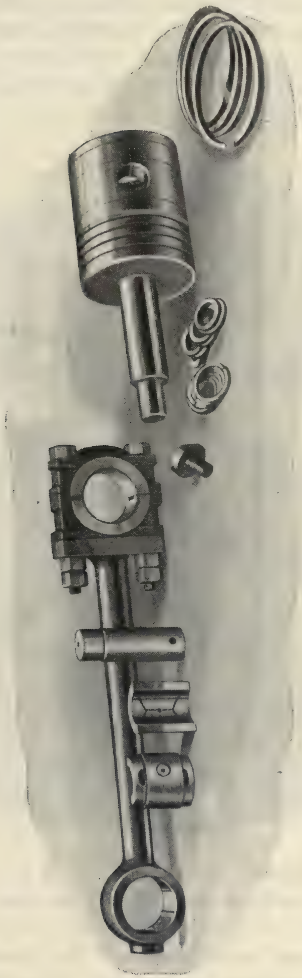
sion with increased output, and the thus augmented heat-transfer, in spite of the temperature remaining the same, calls for observation in the evolvement of Diesel construction. The method of increasing the output is not so much of importance as the process of increasing the pressure. The pressures growing proportionally with increased work and gain in power must be well endurable by the cylinder walls and other sections of the engine, minimizing all stresses.



Cylinder construction to resist tension in addition to bursting strain, is a factor exceedingly vital

Pressures due to the inertia of the reciprocating parts, growing proportional to the square of the number of revolutions, is in itself a problem which requires intimate knowledge. As the working pressures have gained in the same measure, the resulting piston-forces and the tangential-forces as well as the surplus work-areas determining the coefficient for the fluctuation of speed, resulting in increased external strains. As the compression terminal-pressure and the pressure of combustion grow proportional with the increase of output, a higher rate of revolution is to be expected, as far as this is not restricted by the inertia-forces. To judge the conditions of importance in Diesel-operation, the bearing-loads

resulting from the acting piston-forces and the inertia-loads resulting from the inertia-forces; the unbalanced forces and tilting-moments in engines of either the two-cycle or four-cycle type, requires practical



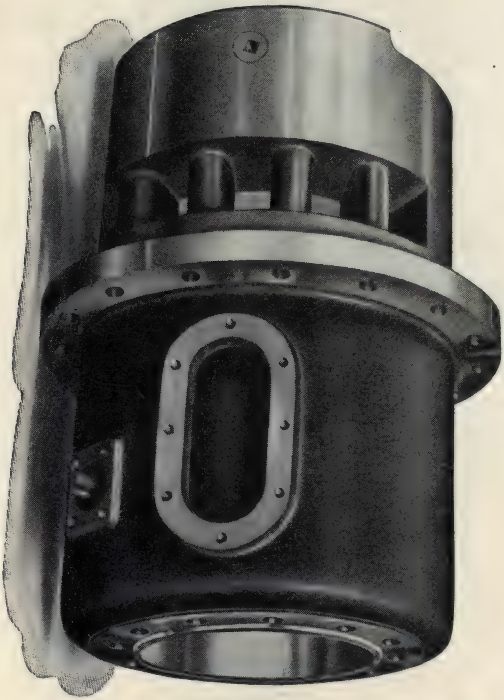
Typical Piston, connecting Rod, Snap Ring, Wrist-Pin and Brasses.
Always keep a spare set of Rings, Brasses, etc., on hand

knowledge combined with theory. The measurements for the mechanism and other vital sections should be computed on the assumption of equal stresses with respect to strength and equal specific uniform-pressures.

SCAVENGING ARRANGEMENTS

The necessity of providing scavenging pumps on two-cycle engines and such problems arising in conjunction with this vital arrangement, in particular on larger types, is a factor which enters in this subject under discussion.

The principal requirement of two-cycle operation demands thorough scavenging of its cylinders of all burned substances. Remainder of these gases will have a tendency to decrease the capacity of the machine. Designers of Busch-Sulzer large two-cycle engines have laid particular stress on this matter of intrinsic importance as will be observed in the description of this respective type.



Cylinder of "Standard" Horizontal Type of Diesel Engine.
A result of careful investigation.

Scavenging comprises two functions—the clearing of the cylinder of the products of the previous combustion, or burnt gases, by means of a current of air, and the supply of the air charge necessary for the next combustion.

A very distinctive feature is the Sulzer Patented Scavenging and Charging System.

Two general methods are employed—head scavenging and port-

scavenging—and one or the other of these, in its older form, is still used on all types of two-cycle Diesel engines, excepting Sulzer's of European construction and the American Busch-Sulzer's.

Head-scavenging—in which the scavenging air enters the cylinder through valve-controlled openings in the cylinder head, and the burnt gases are blown out through piston-controlled ports in the cylinder wall—necessitates the use of one or more comparatively large valves in the cylinder head, rendering the head particularly susceptible to fracture due to heat stresses. This fault has proved so serious, and so entirely unavoidable that there is now a general tendency among builders of two-cycle Diesel engines to discard head-scavenging altogether.

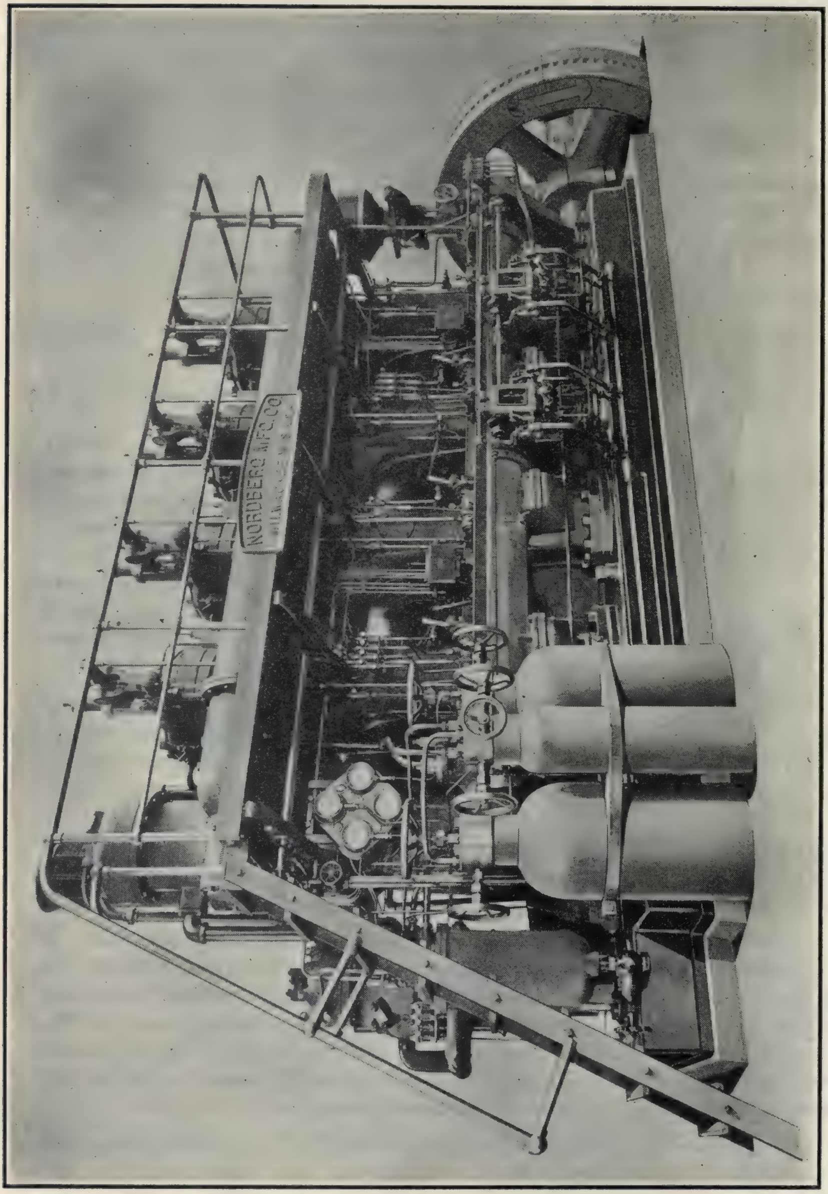
Port-scavenging—in which the scavenging air enters the cylinder through piston-controlled ports in the cylinder wall, and the burnt gases are blown out through similar ports—while it overcomes the cracking of the cylinder head, possesses, in its ordinary form, characteristics which affect the engine detrimentally. It is obvious that the scavenging ports must not be uncovered by the piston until the pressure of the hot gases in the cylinder has fallen below the pressure of the scavenging air; serious scavenging-air receiver explosions have resulted from insufficient attention to this precaution. At full load, the terminal pressure in a Diesel cylinder is about 40 pounds; the scavenging-air pressure rarely exceeds 6 pounds. It is necessary, therefore, that the exhaust ports be uncovered in advance of the scavenging ports, and the amount of this earlier opening is usually about 8 per cent of the piston stroke.

Uncovering the scavenging ports after the exhaust ports, naturally involves covering the exhaust ports after the scavenging ports; the result of which is that, at the time the upwards-traveling piston covers the exhaust ports—namely at about 20 per cent of the stroke—the cylinder is filled with air at very little above atmospheric pressure, which is compressed during the remainder of the stroke. Thus the weight of air compressed, by this system, does not exceed 85 per cent of the weight of a cylinder full at atmospheric pressure.

Moreover, the scavenging by this method is imperfect, and there is an opportunity for burnt gases to blow back into the cylinder before the exhaust ports are closed; the cylinder, therefore, contains somewhat impure air. In very small engines the scavenging may be improved by providing the top of the piston with a projection to guide the stream of scavenging air upwards; but in Diesel engines of even very moderate size a piston of this form would not last many days.

The net result of the foregoing is the inability to perfectly consume, in an engine of this type, the full quantity of fuel which could be consumed if the cylinder contained pure air in an amount equal to the cylinder full at atmospheric pressure.

The Sulzer scavenging system avoids the above described faults in a safe and simple manner. It utilizes port-scavenging; but employs two tiers, instead of only one tier of ports. The piston uncovers the upper



440 B. H. P. Nordberg Engine. Air Compressor and Scavenging Pump at left. Note entire control from floor level

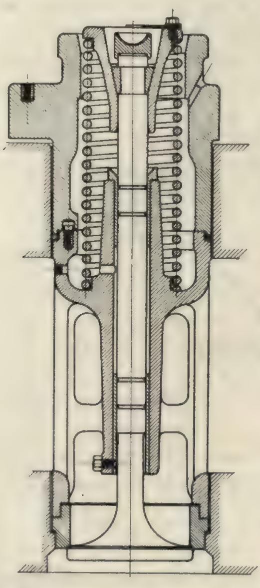
tier of scavenging ports before, and the lower tier after, it uncovers the exhaust ports, but the communication between the interior of the cylinder and the scavenging-air supply or receiver, through the upper ports, is controlled by a timed and mechanically operated valve, which

remains closed until the exhaust ports have been uncovered long enough to reduce the pressure of the gases in the cylinder to nearly atmospheric; after which this valve is opened, while the piston uncovers the lower scavenging ports; a rapid and thorough purging is then effected with complete safety against a blow-back into the scavenging receiver.

Upon its return stroke, the piston first covers the lower scavenging ports and then the exhaust ports; the upper scavenging ports and their valve remains open, enabling the scavenging air to fill the cylinder at full scavenging pressure before the communication is shut off by thc piston. Obviously a blow-back of exhaust gases into the cylinder cannot occur; furthermore, the double tier arrangement and proper form of the scavenging ports insure a clearing out of such thoroughness that substantially no burnt gases remain in the cylinder—analyses have shown that this residue does not exceed 3 per cent. The weight of air compressed is thus substantially 100 per cent of the weight of a cylinder full at atmospheric pressure, and it is possible to perfectly consume the full quantity of fuel.

Incidentally, the effectively directed streams of scavenging air cool the cylinder more evenly than is possible with ordinary port-scavenging.

In accompanying illustration of cross-sectional view of the Carels type of scavenging valve it will be noted that in this case it may be compared to valves of similar construction on four-cycle engines used for exhaust purpose. The method of location of these valves on Nordberg engines in the cylinder head is a feature which has been found satisfactory on this type of construction. The advantage gained may be summarized in, first, added strength to the head itself and, second, establishing more uniform cooling.



Cross-sectional view of Carels type of scavenging valve, used on Nordberg engine.

The importance of establishing a satisfactory scavenging method has caused manufacturers to adopt a double system of scavenging. In the case of double-acting two-cycle engines in some types a scavenging pump for each cylinder was found necessary.

In most marine types of engines the valves are actuated by levers, the cams operated by contact from the cam shaft. The arrangements of providing a double set of scavenging valves, one on each side of the fuel inlet valve, is a feature adopted by some firms.

The principle of the port system, as in the case of the Busch-Sulzer, for its larger types of engines owing to the amount of air necessary in performing the function of scavenging appears to be the future method.

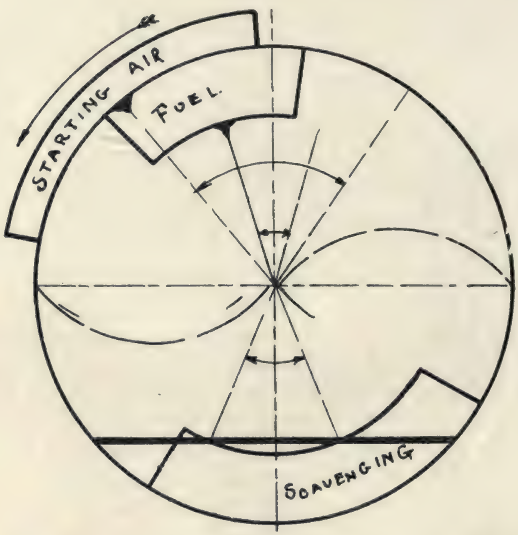
METHODS EMPLOYED IN REVERSING MARINE DIESEL ENGINES

Unlike the Diesel engine for stationary purpose, operating in continual one way direction, the problem of reversing marine engines has been given a great deal of consideration by designers. On late types many new methods have been adopted to accomplish the reversing of engines, which must accurately answer the immediate requirement of giving its motion in either direction for maneuvering.

Unlike a steam engine, following uniform laws of old-established principles, the setting of valves require the most intimate knowledge of problems to be confronted in general operation of Diesel machinery.

Mechanical contrivances imperative to establish automatic operation must be thoroughly understood. Again the varying methods on different machines as will be observed when carefully going over the processes of operation on the different types of engines dealt with in this book, will be found beneficial in studying this prime mover.

In the following illustration a valve-setting diagram for ahead and astern running of a reversible two-cycle engine is shown.



Valve Setting Diagram for reversing of Two-cycle Engine.
(Starboard to Port)

As a matter of fact, in the operation of Internal Combustion Engines, in particular the high compression type, the following proceedings have to be carried out, when reversing is to be accomplished, on most marine engines:

(1) The valve levers have to be lifted off the cams on the camshaft.

(2) The camshaft has to be moved fore and aft to bring the astern cams underneath the rollers of the valve levers, after which the levers must be dropped down again on to the cams.

(3) Compressed air has to be admitted to all cylinders, after this, where perhaps the engine is composed of six cylinders, two have to be placed on fuel and four on air; next, two on air and four on fuel, and finally all on fuel.

If the engine is running when the order is given for it to stop, the fuel supply must be immediately lowered. This is usually accomplished by mechanical arrangement of a hand-wheel turned to the stop position, as indicated on the dial. This causes a partial rotation of a spindle, which raises or lowers the rods. These are attached to sleeves, on which the levers operating the fuel valves are eccentrically mounted. The other end of the lever on the fuel valve cam is, therefore, raised from the cam by this operation and is only brought down on to the cam at the right moment by the movement of the starting wheel. In other words, when the engine is in the stop position the fuel valves and starting air valves are automatically out of operation until the hand wheel is moved.

Assuming the engine is stopped after having been running ahead, and the order is received to go astern, the reversing lever is moved from the back position to the front. This puts compressed air on the motor (some times as in the case of the Vickers types, Servo motors), which by means of a rack motion, first partially rotates the horizontal shaft which lifts the exhaust and inlet valve levers off their cams through the link, then causes the lever to move fore and aft, giving the corresponding motion to the camshaft, after which, by the continued rotation of the shaft and the movement of the link, the valve levers are once more brought down to the cams. Only when this complete movement has been effected is it possible to move the starting wheel.

Immediately the cams are in the astern position this starting wheel is rotated by hand until the indicator on the dial shows that air is being supplied to all cylinders through the distributing valves. The engine then starts up on air, after which, the starting wheel is turned to the next position indicated on the dial, namely, two cylinders on fuel and four on air. This is accomplished by the rotation of the spindle as previously mentioned, allowing the fuel valve levers to come down on their cams. Further rotation of the starting wheel cuts out the air supply and allows four of the six cylinders (taking this method to be on six cylinder engines), and finally all of them, to operate on fuel.

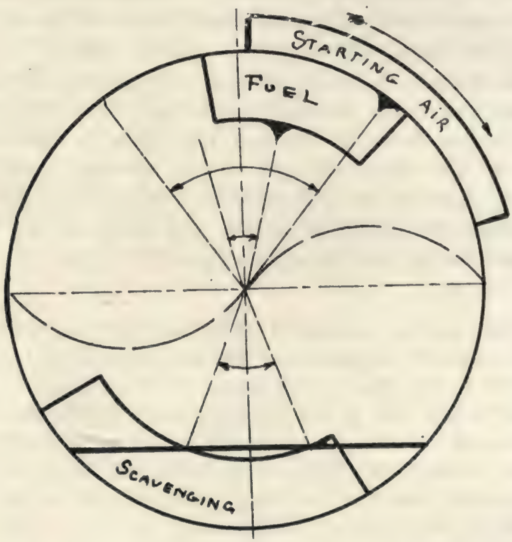
It should be mentioned that the valve levers are lifted off their cams by the movement of the manoeuvering shaft, owing to the fact that these levers are usually mounted eccentrically upon the shaf.

The reason that the fuel valve levers are brought down on to their cams in pairs as described, is that there are cams in this particular

case on the shaft which lift the levers at the time required for putting into action the respective valves, according to the position of the starting wheel.

In many engines hand-pumps are used, operated by levers, in case it is desired to carry out the reversal by hand instead of by compressed air.

In most engines individual cylinders can be cut out by means of hand-levers if the requirement necessitates the same.



Valve Setting Diagram for Reversing of Two-cycle Engine.
(Port to Starboard)

It is the opinion of the writers that the large marine Diesel engines will more and more resemble each other and that certain standards in design will be adopted by all builders, as since long has been the case with the reciprocating steam engine.

The tendency shows already in the general designs of most modern engines. It will be noticed that with the increased demand for larger types of Diesel engines the gradual uniformity in build and general design are identical. It is true, that the tendency of manufacturers adhering to either the two-cycle or the four-cycle type leaves a gap not to be found in steam engine construction, but as both engines of late have been brought to a high stage of perfection, manufacturers in the United States as well as in Europe will find it convenient to establish standards to be adopted governing existing problems in Diesel engine construction.

In most engines of large horse power capacity, the reversing mo-

tions are carried out by the valve-rockers, and like all eccentric movements the action is very peculiar, yet exceedingly simple. In the Werkspoor engine, there are four rockers per cylinder, for the inlet, exhaust, air-starting and fuel valves respectively. Each rocker is mounted on a diagonal-eccentric, the eccentric being secured to the shaft and is free to move in the hub of the rocker. To shift the rocker-rollers from one cam to another it is merely necessary to rotate the rocker-shaft 180 degrees. In the neutral position at 90 degrees the rollers are clear of the cams. The turning of this rocker-shaft requires very little effort, and actually can be done by hand. But, to facilitate the operation, a little double-acting air engine with an oil cushion is provided and this reciprocates a ratchet that is in connection to a ratchet-wheel on the rocker-shaft.

It will be realized, that when the valve-rocker moves from the ahead-cam to the astern-cam the position of its valve-tappet also changes, and this is arranged for by the provision of a double head, or tappet, with an adjusting screw on each. The setting of the rocker-rollers is so arranged that when running "ahead" the roller-face is square on the cam, but in the "astern" position the face of the roller is not absolutely square on the cam, is resting at a slight angle, which is of no consequence because the wear of the astern position is exceedingly slight, partly owing to the very limited periods during which the engine runs astern and partly because of the large size of the roller.

DESCRIPTION OF GOVERNOR AND GENERAL ARRANGEMENT OF CONNECTIONS OF ASPINALL'S GOVERNOR FOR DIESEL ENGINES

The Governor is fitted to a reciprocating lever worked by engine crosshead, or suitable motion, having for preference an angular movement of about 45 degrees, and making about 80 double strokes per minute. The Governor "A" is adjusted to act about 5 per cent. above the running speed of the engines. When the pre-determined speed is reached, the large weight of Governor is left behind on the downward stroke of the special Lever "L" the bottom pawl on Governor carries Engaging Lever "B" into its upper position, which lifts the Suction Valve on Oil Pump off its seat through Rod "J" and Levers "D" and "E." When this action takes place the bulk of the fuel oil, instead of passing through the Delivery Valve to the Cylinder, is returned to the Suction Chamber, and the engines are then slowed down. The amount to which the Suction Valve is lifted off its seat by the Governor is regulated by the Screw "R," which is set so that a small quantity of Oil Fuel will pass through the Delivery Valve to the Cylinder. When the speed of the engines has moderated, the large weight of Governor drops into its lower position, and top pawl depresses Lever "B," which allows the Suction Valve for Fuel Pump to come on its seat; the full supply of

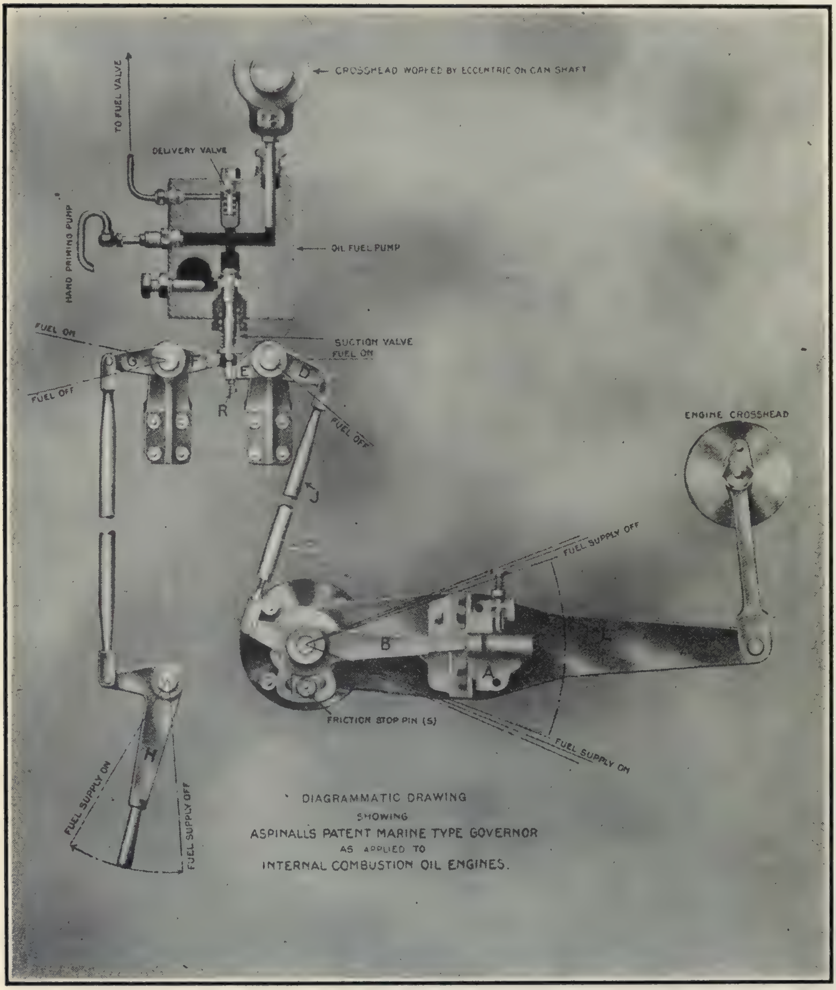


Fig. A. Aspinalls Marine Governor applied to Internal Combustion Engines.

Oil Fuel is then discharged through the Delivery Valve to the Cylinder, and the engines regain their normal speed. The Lever "F," connected with the Hand Gear, is fitted with a Fork End which works outside the Lever "E" of Governor Gear. With this arrangement the Governor Gear, or Hand Gear, work independently of each other. (Note: The arrangement of connections may be varied in numerous ways, providing the principle of the action of the Governor is duly considered.

DETAILED DESCRIPTION OF GOVERNOR

The Governor consists of a hinged Weight "W," operating two Pawls "PP," carried on a frame, which is bolted to a Lever "L," having a suitable reciprocating motion. When the revolutions of the engines are increased by about 5 per cent above the normal running speed the Weight "W" is left behind on the downward stroke of the Lever "L," and reverses the position of the Pawls "PP," causing bottom one to engage with Lever "B," lifting it throughout the upward stroke of Lever "L," and thereby Shuts Off the fuel supply to the engines. On the next downward stroke of the Lever "L" the Detend "Q" is lifted by coming into contact with Lever "B," liberating the Weight "W," and when the revolutions of the engines have moderated, the Weight "W" drops into its lower position and again reverses the position of the Pawls "PP"—the top one now engages with the Lever "B," depressing it throughout the next downward stroke of the Lever "L," and thereby Opens Up the Oil Fuel supply to the engines again.

The Emergency Gear only comes into operation in the event of the engines approaching an excessive speed, such as would occur in the event of loss of propeller or the breaking of a shaft, in which case the Weight "U" is left behind on the downward stroke of the Lever "L" and locks the Weight "W" in the Shutting-Off position, thereby effectually Shutting Off the Fuel supply to the engines. To release the Emergency Gear from locking position press the Weight "W" upwards from the underside, when the Weight "U" will fall out of gear.

INSTRUCTIONS FOR FIXING GOVERNOR

Bolt the Governor on the side of the Lever "L" at a given distance "F" from Fulcrum to Face the Weight "W." Place Lever "L" at Top of its stroke, as shown in dotted lines on Fig. 2, lift the Weight "W" into its upper position, which brings the Bottom Pawl out. Then file out metal from end of Slot in Lever "B" until the engaging end of Lever "B" is ⅜-inch above Pawl, with end of Slot hard up against Stop Pin "S." Then connect up gear between Lever "B" and control gear at fuel pump, which should be in the Shutting-Off position with the gear adjusted so that engaging end of Lever "B" rests on Bottom Pawl. Next disconnect gear between Lever "B" and control gear at fuel pump and place Lever "L" at Bottom of its stroke; lift Detent "Q," which will allow

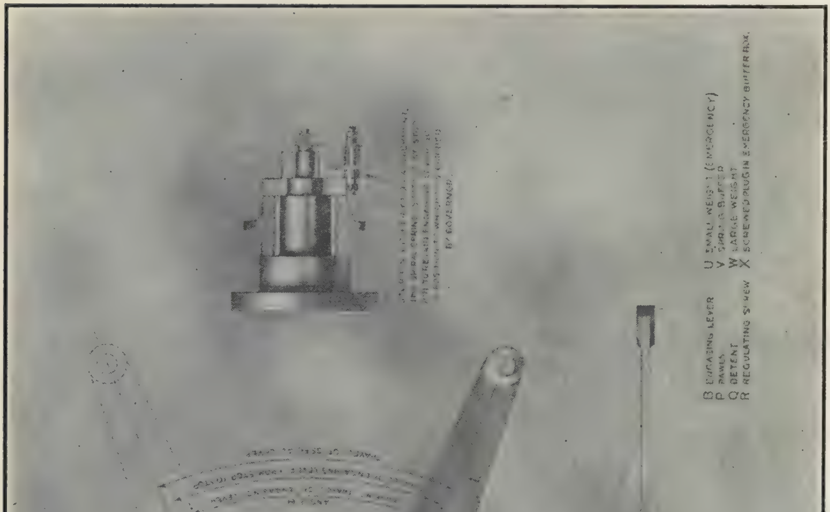


Fig. B. Aspinalls Governor applied to Internl Combustion Engines

Weight "W" to drop into its lower position and bring out Top Pawl. Then file out of other end of Slot in Lever "B" until engaging end of Lever "B" is ⅜-inch below Top Pawl with end of Slot hard up against Stop Pin "S." Again connect up the gear between Lever "B," and control gear at fuel pump, with Top Pawl resting on engaging end of Lever "B" when the control gear, if correctly adjusted, should be in the Opening-Up position.

The distance "F" is fixed by the makers of the Governor after receiving particulars of the engines.

INSTRUCTIONS FOR REGULATING GOVERNOR

To make the Governor more sensitive, i. e., to Shut Off at a less number of revolutions, the Regulating Screw "R" on Spring Buffer Adjustment "V" must be screwed outwards. To make the Governor less sensitive, i. e., to Shut Off at a greater number of revolutions, the Regulating Screw "R" on Spring Buffer Adjustment "V" must be screwed inwards.

To make the Emergency Weight "U" later in its action, take out the Plug "X" and insert a suitable washer inside the Box behind the spiral spring.

HINTS FOR KEEPING GOVERNOR IN WORKING ORDER.

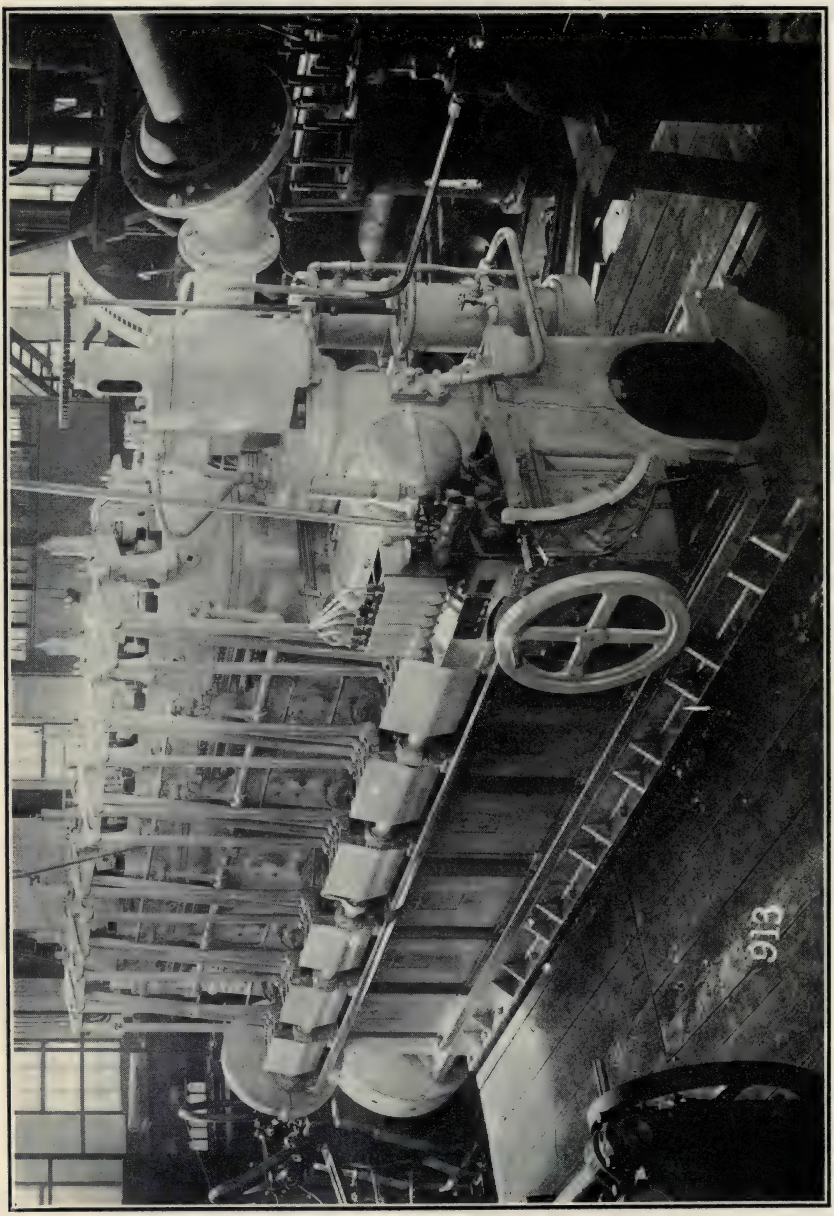
All parts of the Governor should be kept thoroughly clean; a piece of rag and paraffine being used for cleaning purposes. Cotton waste should on no account be used, as particles of same are liable to get into working parts and retard the free working of the Governor. A little mineral or sperm oil should be used for lubricating purposes; oils of a clogging nature to be avoided.

If the Governor is fixed under platform gratings, a piece of canvas or sheet iron should be attached to underside of grating to prevent dirt falling on the Governor. The Weight "W" should be tipped upwards by hand once a day to ensure the Governor and Gear being kept free and in working order.

MARINE DIESEL ENGINES FOR TWIN-SCREW SHIPS.

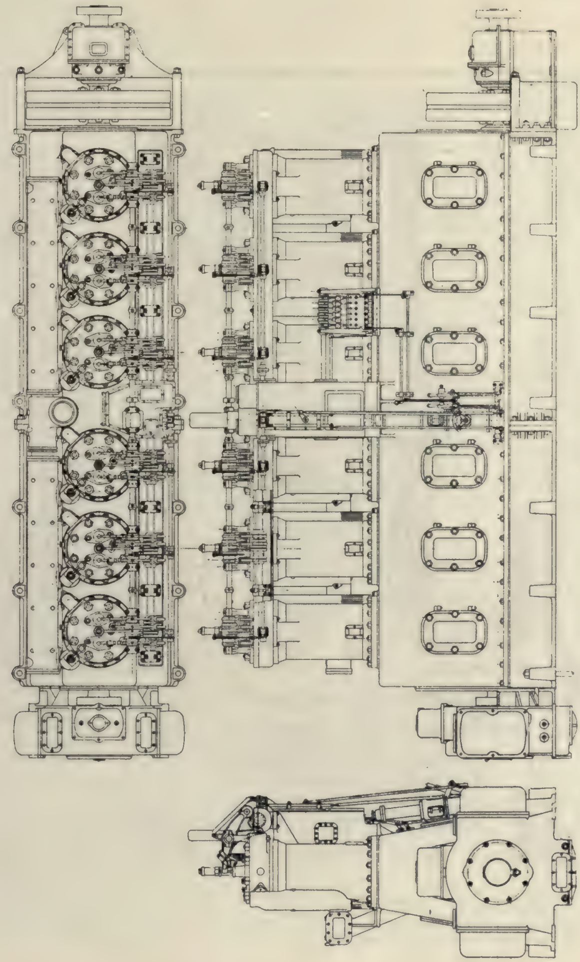
The method of twin-screw propulsion appears to be momentarily far more favored than the single screw process. Firms, foremost in the construction of Diesel machinery claim advantages in favor of the twin-screw propulsion as against the single-screw operation, so commonly found on steam-driven ships. Following reasons are given by the Burmeister & Wain Co.

(1) The engines, shaftings and propellers are lighter than those of the corresponding engine of a single-screw vessel.



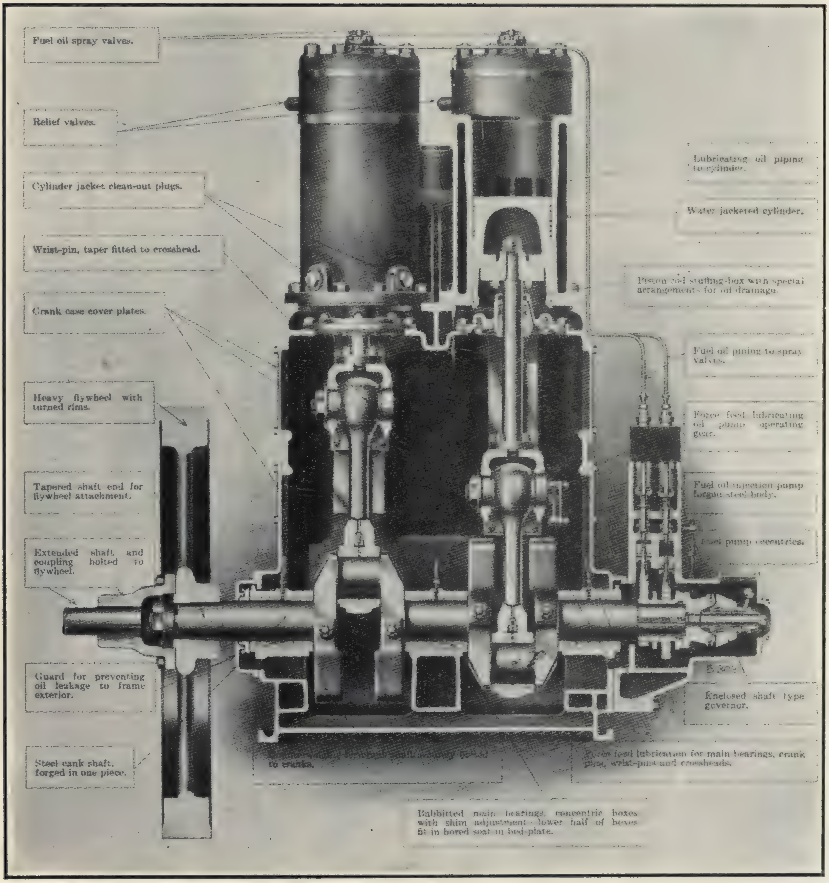
Front View of Engine Installed in M. S. "Carolyn Francis." 390 I. H. P. (300 B. H. P.). McIntosh & Seymour Type.

(2) The engines require less space length-wise, thus the engine room becomes shorter, and in spite of this there is ample room for placing the auxiliary machinery, partly along sides of the ship and partly between the two main engines.



Plan and Side View of Dow Diesel Engine. Records Established With Dow Engines Show Fuel-Economy Exceedingly Low and no Expenses Incurred in Breakdowns During One Year of Operation.

(3) The dimensions of the different parts of the Diesel engines being reduced, they are more easy to handle, and as the whole unit is adapted for forced lubrication, by which all inspection of the working parts, while the engine is running, is rendered unnecessary, if only the pressure of the lubricating oil is kept constant, the attention of the increased number of details will cause no difficulties and no additional labor will be required.



Descriptive View of Worthington Solid Injection Two-Cycle Diesel Engine (Exposed).

(4) The two engines working independently of each other assure a greater reliability, and the efficiency of the two propellers on higher revolutions will be greater than that of a single-screw running at lower revolutions.

The point last mentioned is one of the most essential advantages and has enabled motor vessels to carry through their voyages at such a good mean speed even in bad weather. This is accounted for firstly by the fact that the propelling power is distributed over two propellers thereby attaining a larger thrust pressure. The propellers are placed well clear of the ship's sides, thus assuring a good and free flow to the propellers.

In bad weather these small propellers do not readily get above the surface of the water. Should this happen a better propulsion is nevertheless maintained in comparison with a propeller worked by a steam engine, on account of the following:

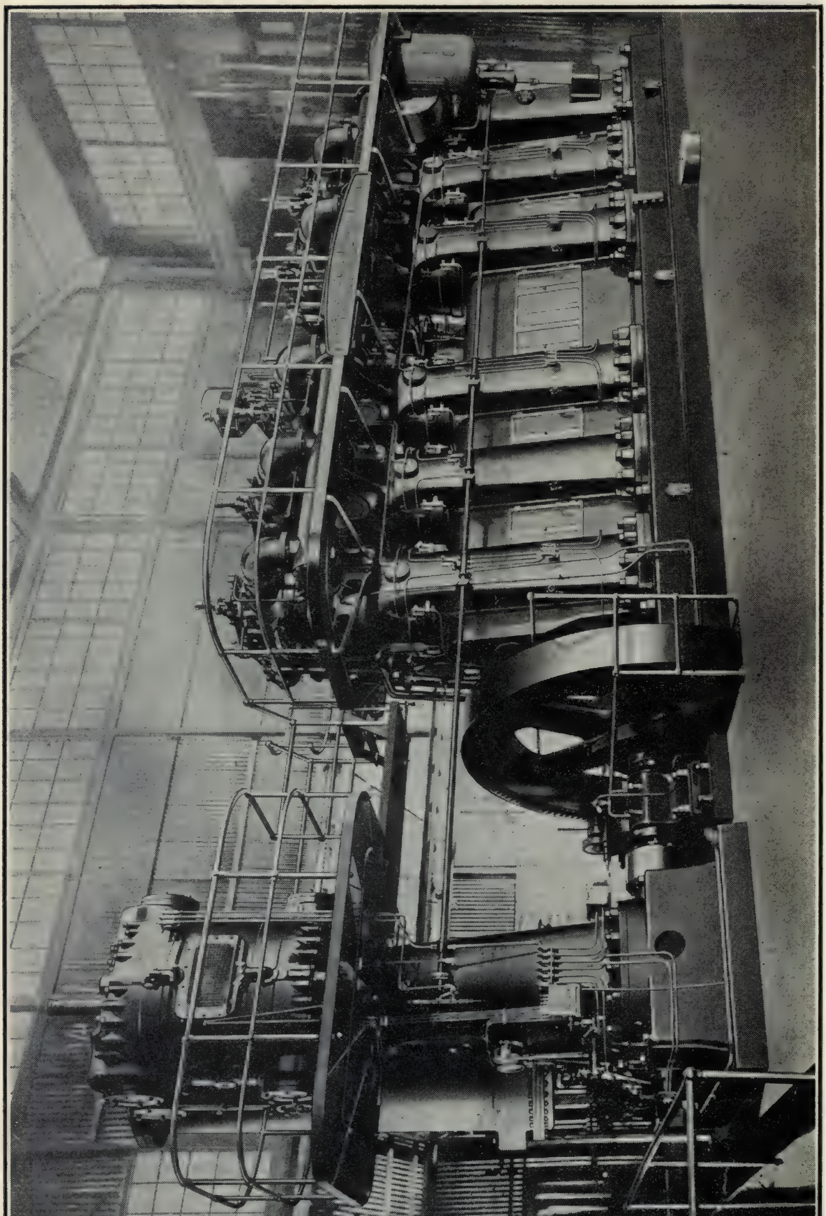
The Diesel engine is furnished with a governor, which as soon as the propellers rise above the surface of the water and the revolutions of the engine thereby are increased by about 10% above the normal, operates and cuts off the oil supply to the cylinders and keeps the engine at about normal revolutions. At the moment the propellers again enter the water and the revolutions are somewhat reduced, the governor acts immediately, adjusting the supply of fuel oil to the normal quantity, so that all the cylinders give full power at once. This is not the case with an ordinary marine steam engine. Even if here the governor cuts off at once for the admission of steam, as soon as the propeller leaves the water, the revolutions of the steam engine will nevertheless be increased owing to the large quantities of steam still remaining in the receivers. When the propeller again enters the water, the governor will at once admit the steam to the high pressure cylinder, but the engine will not be able to attain its normal power, until the receivers again are filled, frequently the steam engine will be practically stopped and the propeller enters the water. In heavy weather the propulsion is therefore better of Diesel-engined vessels than of steamers.

DIESEL HIGH SPEED ENGINES

The problem of high-speed engines for auxiliary purposes enters as one of vital importance in modern Diesel power plants. In many cases the gasoline engine has been found to perform this function on generators, pumps, compressors, etc., in a most satisfactory manner.

Of late high-speed Diesel engines have been built exceeding 400 revolutions per minute. As a matter of fact, on this type of machinery the fuel consumption has been found so low that its competitor in high-speed engines will in future be substituted by high-speed full Diesel machinery.

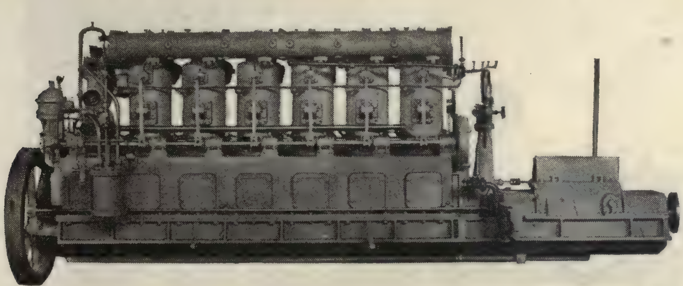
In marine engineering, where it will be not alone a matter of convenience using crude oil for fuel oil solely, in operating the entire unit, but also the importance of space allowance must be considered. In saving



1250 B. H. P. Nordberg Diesel, Direct Connected to Nordberg Two-Stage Air-Compressor at Left.

of engine room space the advantage is with high-speed Diesel machinery. In weight comparison it may safely be said that Diesel driven engines of high speed types are of equal proportion.

The general upkeep of either the two-cycle high-speed or these following the four-cycle principle is a factor which may be worthy of mention. A machine of this kind developed by the Busch-Selzer Company of St. Louis of four-cycle construction shows wonderful results. In this case force-feed lubrication has been adopted throughout. The vertical three-stage compressor, mounted on the end of the engine, is driven off the crankshaft direct from the engine.



A Small Type of Nelesco, Equipped With Paragon Reverse Gear. This Type Is Ideal for Yachting, Fishing Crafts, Etc.

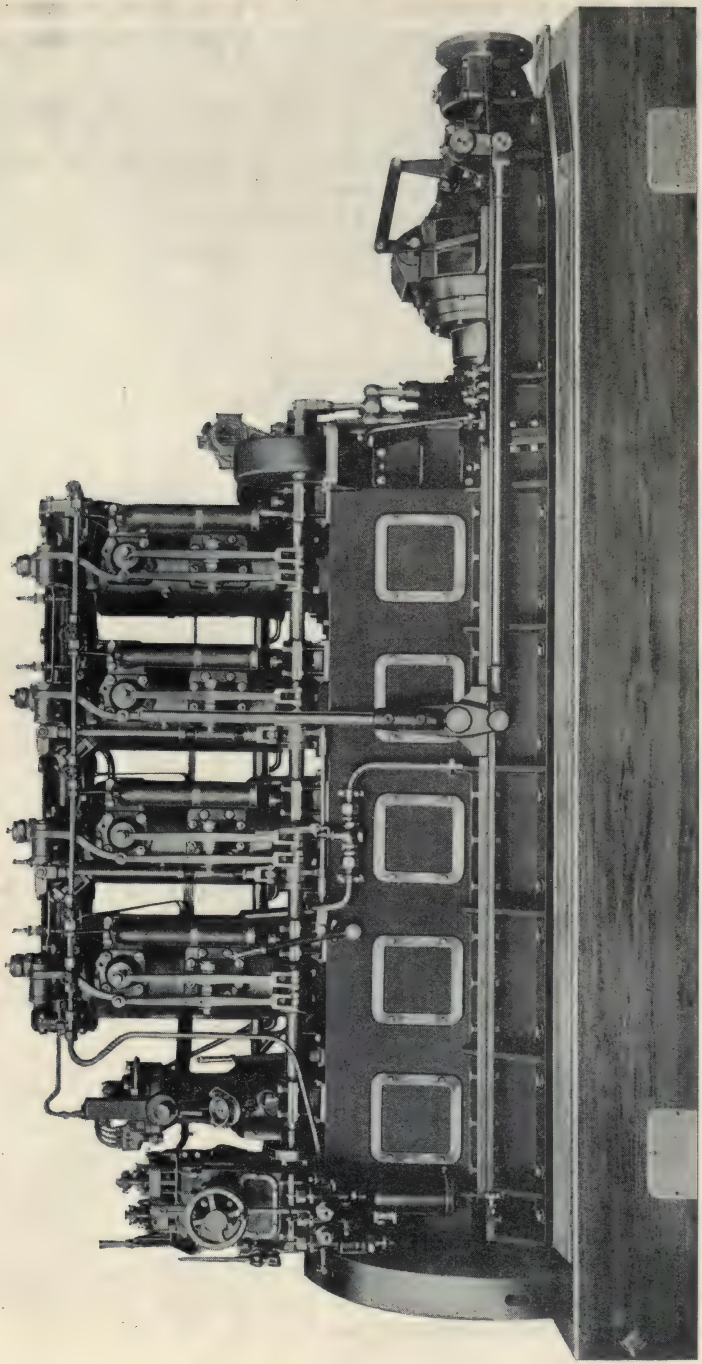
Firms like Krupp, Augsburg-Nurnberg, Steinbecker, Vickers, Werkspoor, Burmeister-Wain, Nordberg Manufacturing Co., have produced engines which are highly commendable for use in auxiliary operation on marine as well as stationary power plants.

It is but natural that the consumption of lubricating oil is higher on high-speed than on low-speed machinery. The amount is such that it not seriously effects operating expenses. It is on a par with the average high-speed gas engine and slightly above steam-driven machines of equal capacities. It may be figured to about .012 to .02 per B. H. P. hour as against .01 to 0.12 lb. fuel oil consumption is in this case about from 3½ to 7 per cent higher, depending upon the design.

ECONOMY IN RIVER-FREIGHTER OPERATION.

Not alone is the Diesel engine a dangerous competitor to vessels operated by steam power, but it outclasses any other prime-mover in economy and efficiency. In the following comparison of Diesel vs. distillate engine on the river freighter "Suison City" of Oakland, California, the impartial result on a trial trip is herewith given:

In this trial trip on the Sacramento River after two 65 h. p. twin-cylinder 12 in. Atlas distillate-engines had been removed and two 55 h. p.



A 120 B. H. P. Nelseco Marine Diesel Engine. The Accessibility Is Notable on This Type.

three-cylinder 8 in. by 10½ in. Atlas-Imperial Diesel engines had been installed in their place, following was the result:

Length over all	84 ft. 5 in.
Breadth	23 ft. 5 in.
Depth	6 ft. 5 in.
Tons, gross	142 tons
Tons, net	73 tons

The following data on the performance of the boat before and after having this change of machinery made is exceedingly interesting as showing in black and white why the Diesel engine, even in small units, must furnish the power in our harbor and coastwise fleets.

Propeller with distillate engines, 48 in. diameter, 44 in. in pitch, 232 R. P. M.

Propeller with Diesel engines, 44 in. diameter, 38 in. in pitch, 340 R. P. M.

Speed of boat with distillate engines, 8 miles per hour.

Speed of boat with Diesel engines, 9.2 miles per hour.

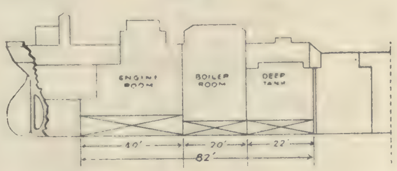
Fuel cost with distillate engines per hour, $3.22.

Fuel cost with Diesel engines per hour, $0.30.

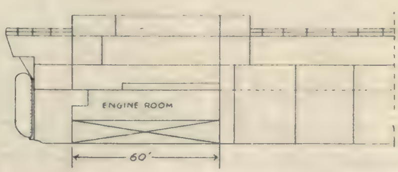
Actual fuel consumption with Diesel engines is 4¼ gals, per hour with two engines and 2⅛ gals. per hour with one engine. Fuel used costs 6 cents per gallon.

This is but one of many conversions from distillate to Diesel engines which the non-availability of distillate fuel and the added economy of the latter type engine has made necessary on vessels of smaller types, as mentioned herein.

(From Motorship, December 1921.)



The Comparison in Space Between Sketch-Drawing of Vessel Equipped With Steam-Power. (Fig. A.)



Sketch Drawing of Vessel Equipped With Diesel Power (Fig. B) Requires no Explanation.

COMPARISON OF EFFICIENCIES OF VARIOUS TYPES OF POWER PLANTS

HEAT UNITS IN FUEL CONSUMED PER BRAKE HORSEPOWER HOUR.

Load Per Cent of Rated H. P.	Simple Non-Condensing Corliss Engines: Engine Rating 100 I.H.P. Boiler Pressure 100 lbs. Steam per I.H.P. Hour, 28 lbs.	Simple Non-Condensing Corliss Engines: Engine Rating 400 I.H.P. Boiler Pressure 150 lbs. Steam Per I.H.P. Hour, 24 lbs.	Compound Condensing Corliss Engines: Engine Rating 200 I.H.P. Boiler Pressure 125 lbs. Vacuum 25 In. Steam per I.H.P. Hour, 15 lbs.	Compound Condensing Corliss Engines: Engine Rating 800 I.H.P. Boiler Pressure 180 lbs. Vacuum 27 In. Steam per I.H.P. Hour, 13 lbs.
100	52,500	41,500	25,500	21,000
75	60,000	47,500	29,000	23,500
50	79,000	59,500	36,500	29,000
25	138,000	99,000	58,000	45,000

Load Per Cent of Rated H.P.	Triple Expansion Steam: Engine Rating 400 I.H.P. Boiler Pressure 150 lbs. Vacuum 27 In. Steam per I.H.P. Hour, 12.5 lbs.	Triple Expansion Steam: Engine Rating 1,000 I.H.P. Boiler Pressure 200 lbs. Superheat 100 degrees Fahr. Vacuum 27.5 In. Steam per I.H.P. Hour 10.5 lbs.	Steam Turbines: Rating 500 K. W. Boiler Pressure 150 lbs. Vacuum 26 In. Steam per K.W. Hour, 21 lbs.	Steam Turbines: Rating 5,000 K. W. Boiler Pressure 200 lbs. Superheat 150 degrees Fahr. Vacuum 28 In. Steam per K.W. Hour, 14 lbs.
100	20,000	17,500	21,000	15,000
75	22,500	20,000	23,500	17,500
50	28,000	24,500	27,000	20,500
25	43,000	36,000	36,000	28,000

DIESEL ENGINES

Load Per Cent of Rated H.P.	Engine Rating 165 B.H.P.	Engine Rating 520 B.H.P.	Engine Rating 2,200 B.H.P.
100	9,000	8,400	8,000
75	9,500	8,900	8,500
50	10,800	9,800	9,000
25	15,400	13,000	12,000

For steam plants add allowance for stand-by, according to character of load.

To obtain equivalent pounds of coal divide B. T. U. by 12,500.

To obtain equivalent pounds of fuel oil divide B. T. U. by 18,800.

To obtain equivalent gallons of fuel oil divide B. T. U. by 143,000.

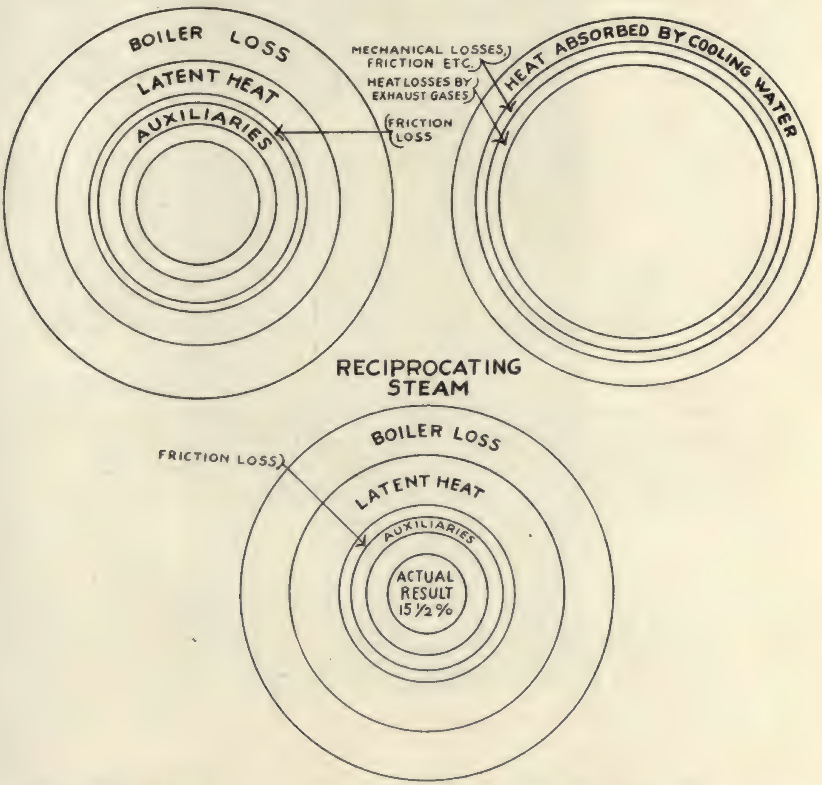
Data furnished by courtesy of Bush-Sulzer Diesel Engine Co., St. Louis, Mo.

COMPARISON IN AVERAGE FUEL COST OF 100 H. P.

Engines driven by Gasoline, Distillate and Engines classified as Diesels.

	Gasoline	Distillate	Diesel
Per hour	2.50	1.70	.42
Per day	25.00	17.00	4.20
Per week	150.00	102.00	25.20
Per month	600.00	408.00	100.00
Per year	7200.00	4896.00	1209.60

Note: In the use of steam it may safely be stated, that a fuel consumption of 300% more than on such machinery where Diesel power is the prime mover, will be required.



Comparison Sketch Between Reciprocating (Steam), Turbine (Steam), and Diesel Power. Upper, left, Turbine; Upper, right, Diesel; Lower cut, Reciprocating Steam.

CHAPTER IX.

AUXILIARY MACHINERY AND ACCESSORIES

OIL PURIFICATION ARRANGEMENTS

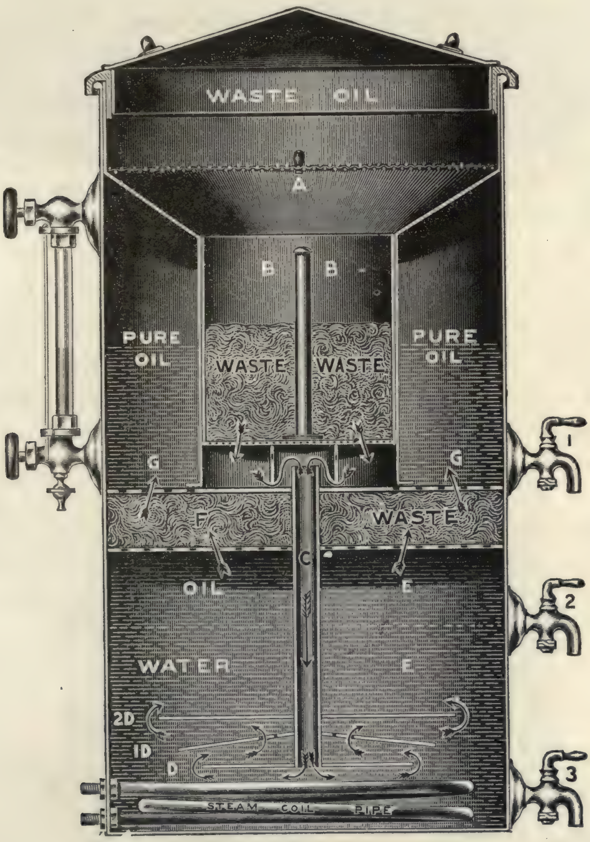
A very important point which should be given careful attention by operating engineers in the operation of Diesel machinery, is the proper maintenance of its oiling and lubrication system.

The tendency of manufacturers, in the design of bearings, has been towards the employment of lubricating oils at much higher temperatures. Such conditions necessitates the use of oils which are suitable for the individual requirements and which must be given such attention and treatment as will minimize trouble in the practical operation of the unit. Experience indicates that the lubricating properties of oil deteriorate under the influence of heat, and when brought into intimate contact with water and air, as occurs under working conditions. These various actions bring about a gradual breaking down of the waxy content into a coagulate or sludge of a brownish color, insoluble at ordinary temperatures in the remaining hydro-carbons, which are the vehicles of the waxy or lubricating agents in solution in the oil. This sludge, with its precipitates, if not removed, adheres to the piping of the oiling system and often obstructs the flow of oil.

The quality of the water which accidentally comes in contact with the oil, due to leakage or sweating of the pipes, affects very materially the formation of sludge. Acids accelerate this action, and alkalies aid in emulsifying the oil. The design of modern Diesels provide for a storage of oil directly in conjunction with the main engine, with pumping, circulating and cooling systems which are independent in operation. Such arrangements ordinarily provide for relatively limited capacities of the oiling system so that it is forced through a rapid cycle of operation. This, in connection with high bearing pressures abnormal temperatures, tends very rapidly to deteriorate the oil, which is not given sufficient time to settle, clarify and purify. The operating practice today consists of removing all the oil after a certain time in service and replacing it with a new batch, the used oil being passed through various types of purifying and filtering equipment, and then stored for future use.

Oil should be selected with careful regard to requirements and conditions of service. Viscosity should be referred not to the conventional standard of 100 degrees Fahrenheit but to the actual temperature at which it will be used, as there is an appreciable variation in the viscosity of all oils, depending on their base and blending, with changes of temperature. The specific gravity of the oil, its emulsifying tendency when

mixed with water, or frothing when churned with air—these actions varying with the temperature of the oil—and, also, to a certain extent, the flash point, are all factors which have an important bearing on the life and service qualities of the oil. They are items which should be regarded as of prime importance in the selection of the proper grade of oil in the case. The oil in service should be tested at frequent intervals because its viscosity and specific gravity and other physical properties change very materially with use and age.



Exposed View of Burt Oil Filter

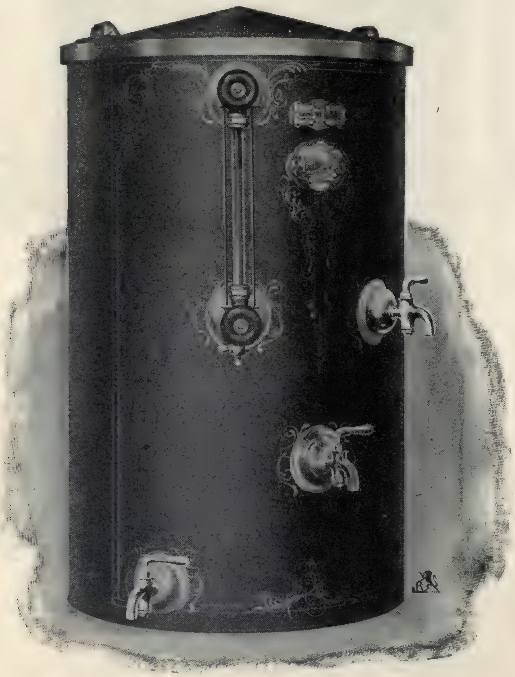
The proximity of the oil tank to the engine arrangements thereto are matters which require serious consideration. The oil tank and connections should be so designed as to prevent, as far as possible, the transmission of any heat from the engine to the oil.

The purification of the oil is, in general, carried out by one of two methods—the continuous system and the batch system. In the first

method the relief valve on the oil system of the engine is made use of to by-pass a portion of the oil continuously into a tank from which it is fed by gravity into the oil purifying equipment and then back into the engine. This acts as a loop in the oil system and keeps the oil constantly in good operating condition.

In the batch system the engine reservoir is completely drained at regular intervals into a tank and a fresh supply of oil is replaced. The used oil is then purified and stored for future service.

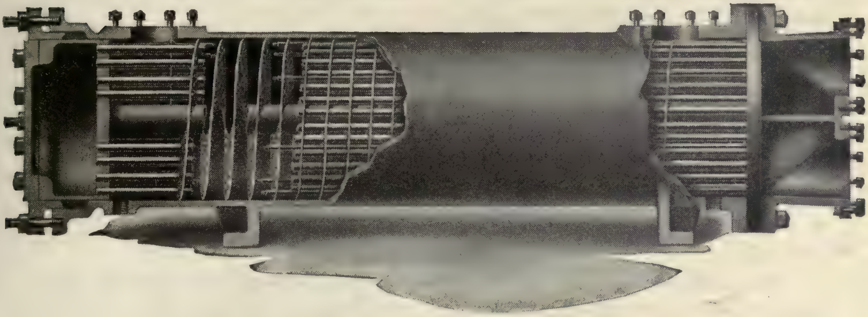
In operating large sized Diesels, it is important to cool the lubricating oil used in bearings, reduction gears, etc., in order that a given quantity of oil can be used over and over again and that the oil supplied to the bearings will be of correct temperature to maintain an oil film of proper viscosity between the bearing surfaces. The Multiwhirl Cooler, manufactured by the Griscom-Russell Company, most efficiently performs this service. The oil is pumped through the shell and the cooling water through the tubes.



Burt Oil Filter, Full View

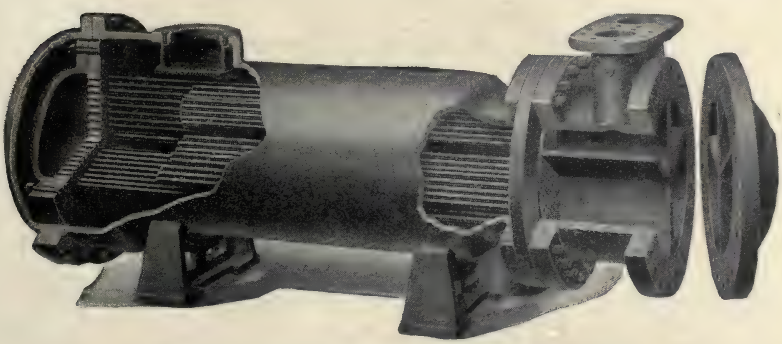
The Multiwhirl Cooler is designed to accomplish the heat exchange between any two liquids, one or both of which may be water; the condensing and subcooling of any vapor or any similar service. In actual

practice it has proven to be an indispensable equipment in such plants where Diesel power is used. The advantages of this oil cooler may be summarized in following:



Multiwhirl Oil Cooler—Exposed View

1. Helical baffle—long smooth oil path—minimum pressure drop.
2. Tube bundle removable, facilitating inspection and cleaning.
3. Tubes expanded into tube plates; no sweated joints.
4. Floating head construction; no expansion strains on tube joints.
5. Outside packed head; this construction eliminates any possible leakage of water into oil through faulty packing.
6. Compactness of unit; this is permitted by the high rate of heat transfer secured in the Multiwhirl Cooler.
7. Installation in any position; the Multiwhirl Cooler may be installed in any position with equal efficiency if liquids (not vapor) are being handled.



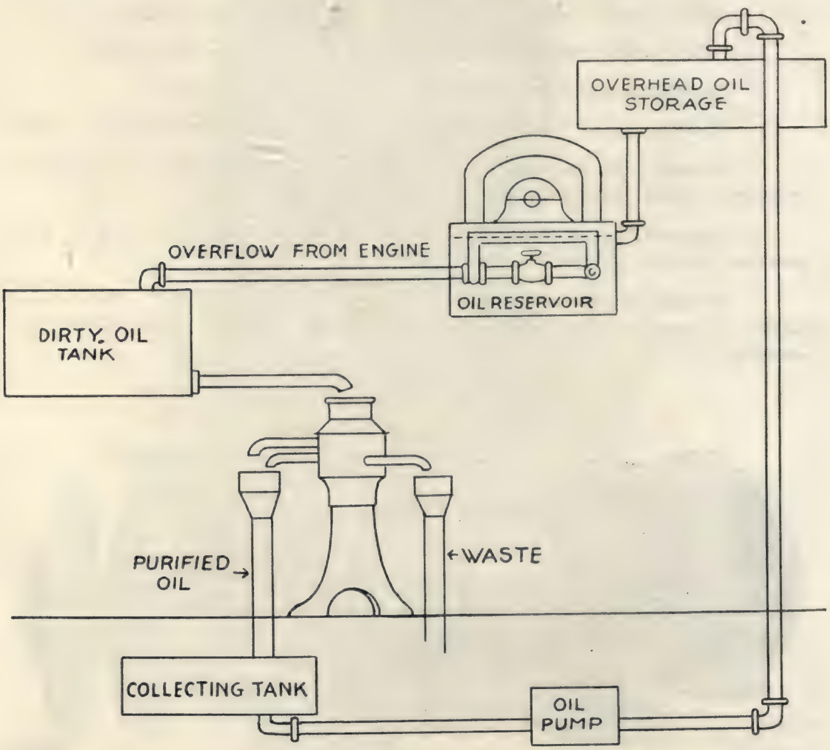
Griscom-Russell's "G-R" Instantaneous Heater

To insure proper operation of valves, such as fuel-inlet valve, etc., it is imperative that certain grades of oil, in particular those of low viscosity, be heated. In accompanying illustration of the Reilly Oil heater a good view is allowed on its interior construction. The oil is circulated through the coils and is heated by high pressure of steam application passing on the interior of the shell.

Internal joints or flanges are to be avoided in oil heater construction to prevent leakage of oil into steam space of heater. Such oil would eventually reach the engine and owing to the fact that water in oil has a tendency to restrict the efficiency in power production it would give additional loss in generating.

Water in lubricating oil should be eliminated as much as possible. It will create heat and will be found detrimental in general purpose of cooling.

To engineers of steam plants, where oil is used on boilers, the necessity for equipment in oil purification, pre-heating of oil or cooling on turbine-driven machinery is imperative. In following illustrations a few apparatus are shown adapted today in plants where Diesel power is used.



The Equipment of De Laval's Oil Separators Assures an Excellent Method of Oil Purification

In figure (a) the illustration shows the Wheeler type of pre-heating of oil. It will be noticed that this arrangement consists of essentially an enclosed cylindrical vessel. Small tubes directing steam through it which causes the heating of oil to be accomplished by coming in contact with the surrounding oil.

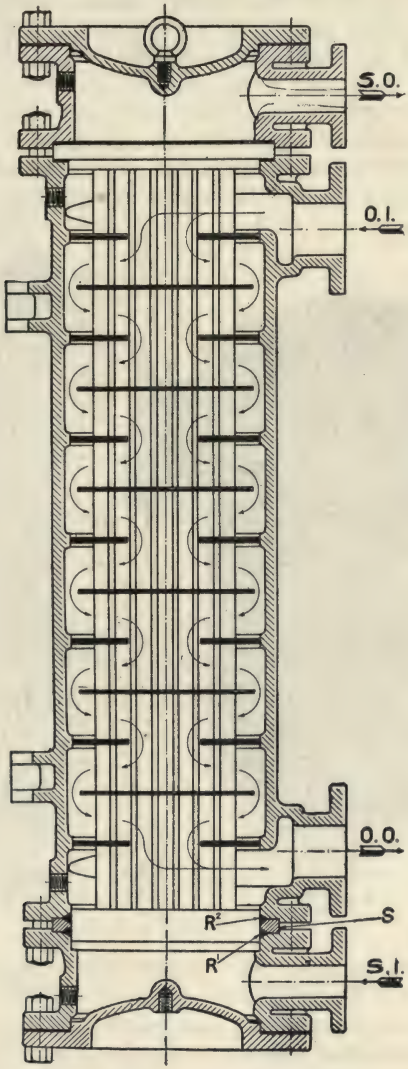


Figure (a). Diagramatical View of Wheeler Type of Oil Preheaters

Figure (b) shows the type of the Elliott Company of Pittsburgh, Pa., known as the Welderon Receiver Separator. Its function is to separate impurities which may cause the clogging up of fuel valves and the piping

system. The Welderon Receivers are of standard construction. The oil and entrainment are prevented from passing into the outlet by baffles consisting of a double row of V-shaped plates. The separator is fitted with a manhole through which a workman can enter the separator and clean the baffles in place.

To be efficient in the removal of sediment from oil it is essential that the baffle plates in an oil separator be cleaned at frequent intervals to prevent their becoming gummed. The flanges are threaded, then welded to the ends of the through-pipe by a special process and form the nozzles of the separators. The body of the receiver is also welded to the through-pipe, which makes an absolutely tight joint. All other joints and seams are riveted, but inasmuch as the through-pipe relieves the receiver of all strains, due to vibration in the pipe line or to contraction and expansion due to change in temperature, the riveted construction can safely be used.

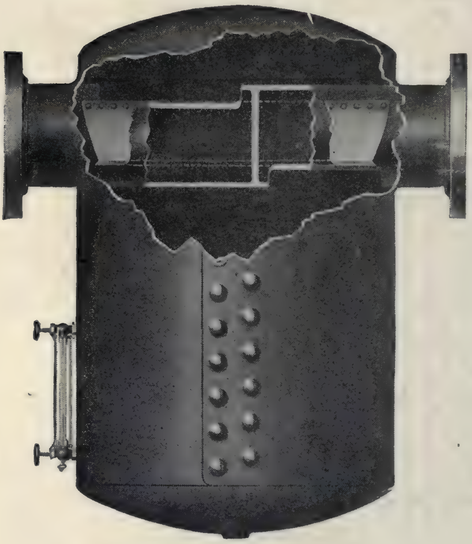
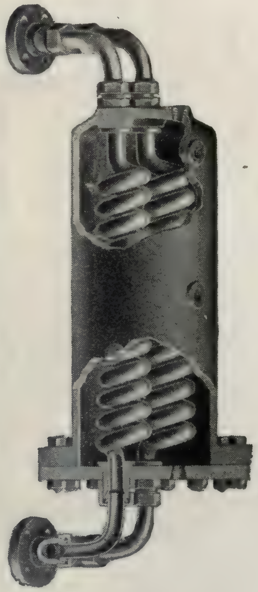


Fig. (b). Welderon Receiver Separator



Reilly Oil Heater—Exposed View

The illustration in figure (d) represents the "Bundy" Oil Separator, manufactured by the Griscom-Russell Company, Massilon, Ohio.

Instead of a single separating plate the Bundy Separator has a number of such plates, thereby insuring that any oil which passes by the first or second plate will be caught by the plates which follow. These plates are of the grid type, the grids being so constructed that the columns of adjacent grids are staggered.

The surfaces of baffle plates used in oil separators are usually left unfinished as the oil will cling to this rough surface more readily than to a finished surface. If a separator actually separates, some portion of the constant stream of oil passing over the separating surfaces will adhere and bake on, thus gradually coating the surfaces and impairing the efficiency of the separator. This result accompanies real separation in any type of separator. Cleaning is therefore necessary if the efficiency of the separator is to be maintained. In order to permit the cleaning of the Bundy plates, a door in either the side or top of the main casting permits their easy removal as each of these plates is a separate casting and they are not attached in any way to each other or to the main separator casting.

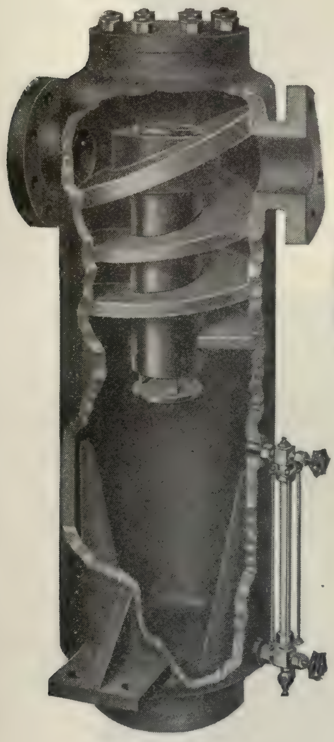


Fig. (C)
Stratton Oil Separator
Horizontal Type (Exposed)

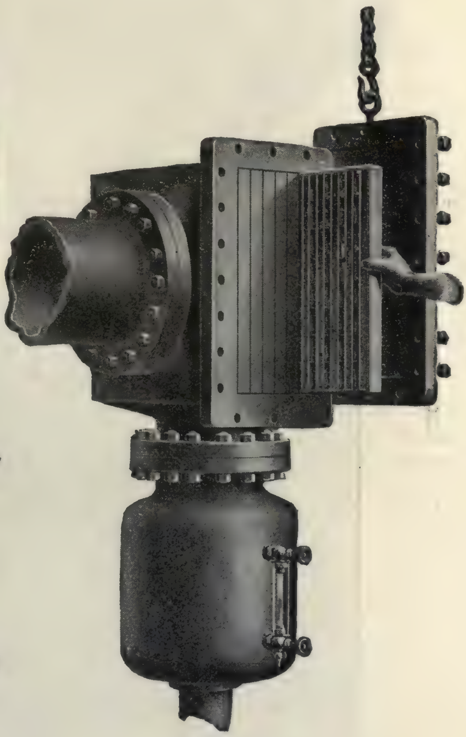
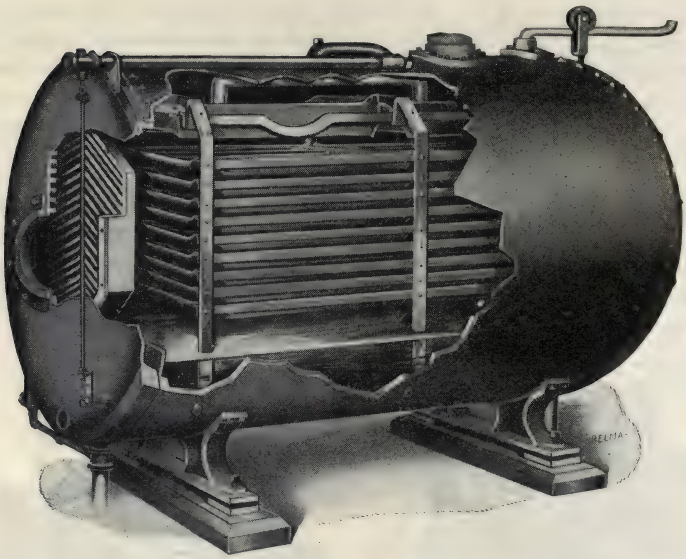
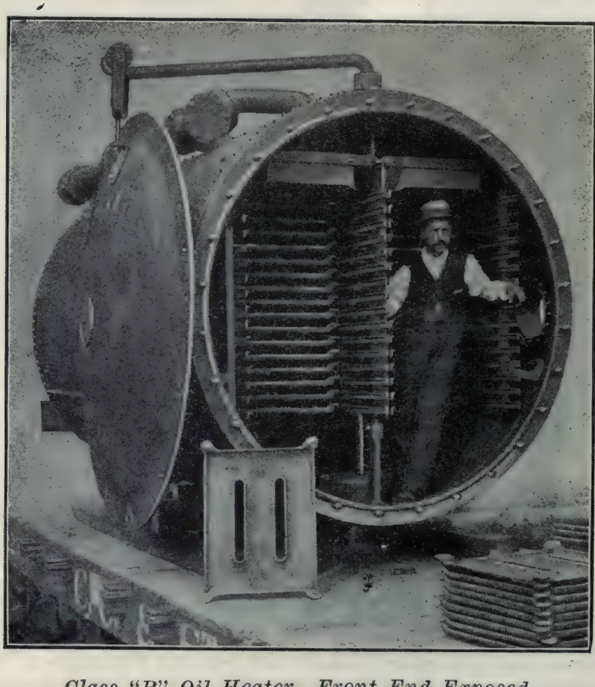


Fig. (D)
Griscom-Russell's
"Bundy" Oil Separator

The separator plates should be thoroughly boiled for about 6 to 12 hours in a strong solution of potash, or soda and water. Their cleanliness will be detected by rust appearing on the surface when dry. After cleaning, the plates can be replaced in the separator with their rough cast



Hoppes Mfg. Co.'s Class "R" Oil Heaters, Showing Multi-Trough Shape I Pan



Class "R" Oil Heater—Front End Exposed

iron separating surface restored to their original condition. The best way to determine just when the cleaning should be done is by trial and observation. On old work, where the piping has become foul and coated through long usage, it is reasonable to suppose that the Bundy separator will require closer watching than where new piping is used.

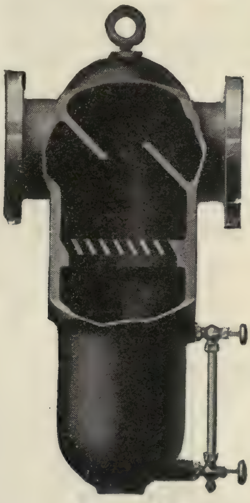


Fig. (e). A "CL" Oil Separator

In the illustration (f) of the multiscreen Filter, a re-design of the well-known Reilley Multiscreen Filter and Grease Extractor is shown. This installation is adapted for ships on over-seas voyages. In particular where steam is used in conjunction with Diesel power.

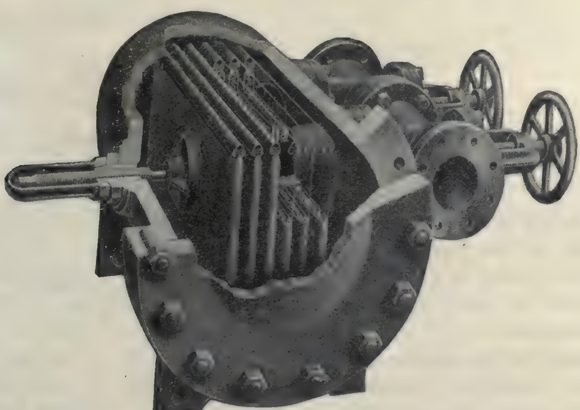
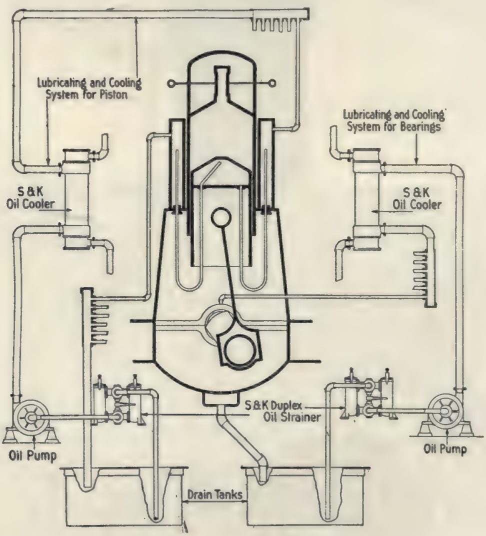


Fig. (f). Griscom Russell's "G-R" Multiscreen Filter

HELPFUL HINTS ON LUBRICATION AND LUBRICATING OIL

If the oil supplied to a bearing is relatively cold, the turbulence is not vigorous, and consequently a particle of oil that is in contact with the bearing and that has absorbed its share of the heat, does not move away fast enough and therefore is not rapidly replaced by another colder particle of oil. The result is that the greater part of the circulating oil during its travel through the bearing does not come into contact with the metal surfaces but passes along without absorbing any heat. Under these conditions, the bearing becomes heated to a point where the temperature difference between it and the oil film in direct contact is great enough to transfer the heat by conduction from the bearing to the oil film. This comparatively small portion of hot oil intermixes with the remaining larger quantity of cold oil, establishing at the outlet of the bearing a low oil temperature which by no means indicates the temperature of the bearing itself.



Oil Cooling and Lubricating System for Internal Combustion Engines by Schutte & Koerting's Method

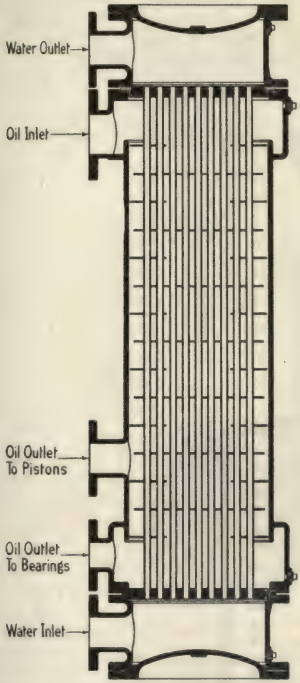
The temperature of the oil may be surprisingly low, whereas the temperature of the bearing itself may have reached the allowable maximum. The bearing is kept cool, not by establishing a low outlet oil temperature, but by bringing into contact with the surfaces of the bearing as many particles of oil per unit of time as possible. The best practice is to circulate the oil energetically and to recool it to a temperature below the normal operating temperature of the bearing. According to tests conducted by the General Electric Company, this bearing temperature is about 160° Fahrenheit.

In the illustration of the Schutte & Koerting Oil Circuit an ideal System of proper cooling is shown. It is a mistake to refer the outlet temperature of the oil to the outlet temperature of the cooling water. The particular oil temperature at which the Diesel engine operates most efficiently should be maintained irrespective of cooling water temperature. The latter can be regulated at will by controlling the amount of water passing through the cooler.

In a test on a standard No. 7 cooler, in which 60 gallons of heavy Texaco Ursa oil were to be cooled per minute with 160 gallons of cooling water per minute, the results given in the following table were obtained.

TABLE OF TEST OF No. 7 SHUTTE & KOERTING COOLER

Inlet Temp. of Water, Deg. Fahrenheit.	Inlet Temp. of Oil, Deg. Fahrenheit.	Outlet Temp. of Oil, Deg. Fahrenheit.	Temp. of Drop of Oil, Deg. Fahrenheit.
85	156	118	38
70	140	103	37
60	128	93	35
50	116	83	33
40	104	73	31



Sectional Elevation of Oil Cooler of the Schutte & Koerting type, for Re-cooling Lubricating Oil and Cooling Oil from Diesel Engine Pistons and Bearings.

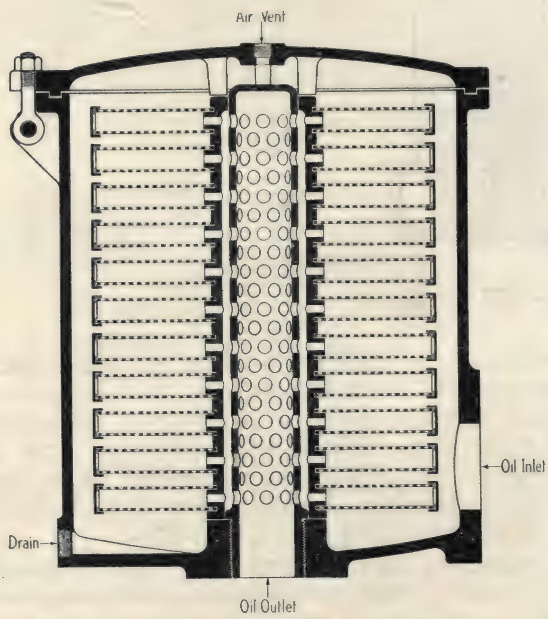
In all instances, the difference between the water-inlet and oil-outlet temperatures was 33° F. Furthermore, the temperature drop of the oil decreased with a decrease in the initial oil temperature. This is primarily due to the fact that as the temperature of the oil diminishes, the oil becomes thicker and more viscous, and its movement along the cooling surfaces sluggish. A more or less stagnant layer or film of oil that forms also retards the cooling materially.

A large quantity of oil circulating at a small temperature drop has been found to give better results than a small amount of oil at a large temperature drop. The oil should not be passed through the bearings at too low a temperature, since cold oil, because of its greater vis-

cosity does not absorb the heat of a bearing as well as oil at a higher temperature. The Schutte & Koerting Company recommends for stationary service an oil temperature drop of 30° F., namely, from 150 to 120° F., since from the experience of most engine builders, this range gives the best operating results. Under these conditions a high rate of oil flow in the bearings is provided, as well as a thin oil film that insures a vigorous movement and rapid efficient absorption of heat in the bearings.

Table, Showing Variation of Specific Heat of Texaco Ursa Oil With Temperature.

Temperature of Oil Degree Fahrenheit.	Specific Heat of Oil B. T. U. per lb. per Degree Fahr.
100	0.437
120	0.450
140	0.462
160	0.474
180	0.486
200	0.497
220	0.508
240	0.518
260	0.527

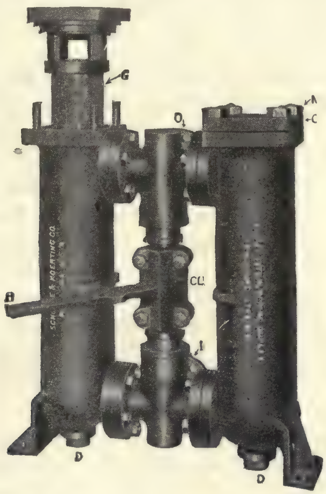


Sectional Elevation of Lubricating Oil Filter for Diesel Engines

If the temperature of the oil entering the cooler is comparatively low, the cooler will necessarily be of larger dimensions to accommodate the larger quantity of water necessary to absorb the same amount of heat.

The oil best suited for Diesel engine lubrication is one that will withstand contamination with impurities under the conditions of most severe service,—high surface speed, rapid circulation of oil, presence of water, etc. With poor grades of oil, the lubrication is inefficient, and due to the heat generated, bearing temperatures are high, the oil is oxidized, and its life limited a few months.

Excessive oil temperatures are sometimes due to the proximity of the oil drain tank to the close distance of engine. Frequently the lubricating oil contains entrained particles of air in the form of bubbles. When the circulation is rapid, these bubbles burst and scatter the oil in fine sprays or vapor.



Schutte & Koerting's Duplex Oil Strainer

This vapor must be removed from the bearing housing since it is likely, where in case of engine being used for electrical generation purposes, to creep into the electric generator, where it can cause trouble For this purpose the oil pipe is made sufficiently large; otherwise an additional pipe is used, venting the oil reservoir of the bearings at a higher level than the original return pipe from the bearing housing. Foaming occurs whenever large quantities of fresh oil are added to the lubricating system. This, however, will disappear after a few hours operation. Should it persist, air is probably being sucked into the system through the oil pump. All air leaks should be eliminated as quickly as possible.

Solid impurities, such as particles of dust and dirt, iron oxide, etc., contaminate, and break down some grades of oil quickly. Such oil becomes dark in color, its viscosity is high, and it forms a sludge that is deposited in the system, usually in the oil cooler. If, furthermore, a slight trace of water is present, the oil will emulsify considerably.

In instances where a new engine is started up for the first time, there will always be in the lubricating oil particles of core sand, cotton waste, scale, etc. Therefore, after the engine has been in operation for two or three weeks, the oil should be removed, put into a settling tank, where the foreign substances will deposite at the bottom, and then the good oil separated off from the top. This oil can be subsequently used as make-up oil.

If it is found that the percentage of impurities is large, the oil should first be heated, separated as before, and then run through a filter before being introduced into the lubricating system. A more definite and positive means of removing solid impurities and foreign substances from the oil is provided by the Duplex Oil strainer of which accompanying illustration gives a view.

The Duplex Oil Strainer removes, dirt, sediment, and any foreign material that has accidentally gotten into the oil. It is of sufficient capacity so that one side may be cut out for cleaning purposes without interrupting the flow of oil through the other side, that is, one side of the strainer can be cleaned while the apparatus is running under full power. It is operated with a single lever. It has a free area through the straining screens that permits of long usuage without causing the system to become choked and clogged. Sometimes the strainer is by-passed,—the by-pass being provided with a relief valve so adjusted that it opens when a fixed differential pressure occurs across the strainer. In this way the oil flow is not interrupted; but an alarm must be provided to indicate when the valve opens, otherwise the clogged strainer will go unnoticed.

When water is present, the result is an emulsification of the oil, With large quantities of water, the mixture assumes a dark yellow color. If a sample of the mixture is removed, and heated in a test tube, it will separate out into oil at the top, milky water in the center, and a slimy sludge at the bottom of the tube. The oil is darker in color, and somewhat heavier than the original oil. It also has a strong characteristic odor.

Water is the cause of considerable trouble, since in appreciable quantities, it forms a sludge that clogs up the passages of the closed lubricating system, and causes the temperature of the circulating water to rise. This condition is usually an indication that insufficient oil is brought to the bearings for cooling purposes. Frequently the engine must be shut down in order to cleanse the system thoroughly.

When water leakage is unavoidable, it is good practice to put a water drain into the bottom of the drain tank. This water drain should be opened once in every 24 hours, so that any water accumulating in the

system can be withdrawn. The drain cock should be left open until clear oil appears. It is also advisable to open the drain cock before starting the engine. The suction of the oil pump in the drain tank should be at least two to four inches above the bottom of the tank, so that any water which may have separated out will not be mixed with the oil, and passed into the lubricating system. In some engine rooms the usual procedure is to remove each day from the bottom of the oil tank three to six gallons of oil. This is heated in a separating tank, and later filtered.

It is advantageous to have a large quantity of oil in circulation, with large oil tanks in which the oil has time to rest and to separate from the air, water, dirt and other impurities collected in the system. There is always more or less air in the circulating oil. At about 180° F., the air oxidizes the oil, causing it to assume a dark color. Carbon is deposited and frequently chokes up the oil inlet to the bearings, and causes the oil in the governing gear to stick. In an efficient oiling system in which there is no waste or leakage of the oil, and little or no water, the amount of make-up to be added every week is very small.

Where poor grades of oil are used, the addition of new oil throws down a dark deposit throughout the entire system. This is particularly true with the heavier grades of oil. It always pays to use the proper high grade oils, since these separate quickly from impurities. They reduce friction to a minimum, prevent high bearing temperatures, and insure correct lubrication and efficient operation.

RECOOLING JACKET WATER OF INTERNAL COMBUSTION ENGINES

On board ships or in plants where the supply of fresh water for cooling the jackets of internal combustion engines is insufficient, it is necessary to use the same clean jacket water over and over again, and for this reason, to recool water.

This can be done to advantage in the water cooler, wherein the same principles of construction are employed as in the oil cooler, as shown in the illustration of the same, and the same exceptionally high heat transfer, low weight and small space requirements are obtained.

Since there is no necessity of replenishing the jacket water, the same water is used over and over again. Thus, due to the continual circulation, the possibility that any sediment in the water will settle in the passes of the cooling packet is reduced to a minimum, and cleaning of the jackets is unnecessary. All clogging of the passes is avoided, as are strains and cracked cylinders caused by uneven distribution of heat.

By using a recooler, the cost of fresh water is reduced or entirely eliminated. Furthermore, the heat coming from the heater can be used to advantage in many ways, increasing materially the economy of the engine plant. The cooling water in the cooler must be capable of carrying off from an ordinary commercial or even a naval type of Diesel engine an amount of heat (including that abstratced from the lubricating oil), equal to about 35 per cent of the total heat in the fuel consumed.

For example: An engine consuming per B. H. P. hr. (horsepower hour actually delivered at the coupling end of the crank shaft), 0.40 lb. of fuel oil of 18,000 B. T. U. per lb. or 7200 B. T. U. per B. H. P. hr., will require sufficient cooling water to carry off 2520 B. T. U. per B. H. P. hour.

If the water has a temperature of 70° Fahrenheit, and the discharge temperature is limited to 100° Fahr., the quantity of cooling water required will be 84 lbs. or about 10 gallons per hour per B. H. P. Under these conditions the water pressure will be less than 30 lbs. per square inch.

If the total heat of the engine amounts to about 775,000 B. T. U. per hour, about 260,000 B. T. U. per hour would be transformed into mechanical work. This means a heat loss (total heat loss = total heat available — heat transformed into mechanical work) of 775,000 — 260,000, or 515,000 B. T. U. Of this amount, about one-half must be absorbed by the jacket water. One half equals 257,000 B. T. U. per hour, the amount of heat to be absorbed by the jacket water, and to be surrendered to the water in the cooler.

If 250 B. T. U. are to be transferred per hr. per sq. ft. of heating surface per 1° Fahr. mean temperature difference between the hot and cold water, and if the hot water is circulated around the tubes at a velocity of 18 in. per second, then the velocity of the cooling water through the tubes must be about 19 in. per second. For a heat transfer of 275 B. T. U. this velocity must be 23 in. per second.

RECOOLING JACKET WATER BY MEANS OF AIR

The Water Cooler, used for recooling the jacket water of internal combustion engines by means of air, consists of a bundle of straight oval or round tubes. The ends are cast into the tube sheets. The jacket water to be cooled passes through the tubes, and air is blown across the tube surfaces by means of a blower.

The small weight and space occupied by the apparatus make it indispensable for this purpose. Oval or round tubes are in staggered formation. Thus the air is split into numerous fine streams and comes into thorough contact with the tube surfaces. The frictional resistance through the cooler is small. The use of oval tubes insures high heat transmission, since the water column flowing through the tube is of oval cross section and every particle of water is close enough to the interior tube surface to be able to surrender its heat effectively and completely. There is no dead central core of water as in round tubes of the same capacity.

The design of the recooling plant must be based on the maximum heat that is to be absorbed from the engine by the jacket water, and surrendered to the air in the cooler. Furthermore, the power required by the blower should remain within reasonable limits,—at the utmost

not more than 5 per cent of the engine capacity,—the air should be drawn through the cooler in the correct manner.

In a general way, it may be stated that there must be a temperature difference between the surrounding air and the water leaving the cooler of at least 30° Fahr., if the cooling plant is not to be abnormally large and expensive. For example, if a recooling plant is to be operated with air at 80° Fahr., the water would not be cooled below 115° Fahr. or 110° Fahr. at the lowest.

Schutte & Koerting Water Coolers are built in sizes adapted for engine capacities of 5 H. P. up to large units of 200 H. P. For still larger units several coolers are used in combination.

INTER AND AFTER-COOLERS FOR AIR COMPRESSORS.

The air cooler is extensively used in connection with air compressors for cooling the compressed air. The apparatus is made to withstand any pressure required, and is designed so as always to retain the advantages of a very compact arrangement. Round tubes are employed and the tube bundles are inserted in cast iron or sheet iron casings.

Both oil and water eliminaters are provided, thus practically all of the entrained moisture and oil is removed. This is essential, particularly when the air is subsequently used in air agitators around machinery where the inter and after-cooling methods are imperative. Also by insuring the removal of all entrained moisture, the possibility of freezing in winter is entirely eliminated.

The air is carried around the outside of the tubes, and the cooling water through the tubes. As it is a simple matter to clean the insides of the tubes, very dirty water can be used. Notwithstanding the small dimensions of the apparatus, the resistance to flow is low, and large quantities of air can therefore be handled with small frictional losses.

The operation of a two-cylinder double-acting two-stage air compressor with an inter-cooler and an after-cooler is as follows:

Air enters the cylinder through the open inlet valve, which, at the end of the suction stroke, closes; as the motion of the piston compresses the air filling the cylinder, the discharge valve opens automatically, and allows the compressed air to pass the inter-cooler, where it is cooled, and thence discharged to the second cylinder or to the subsequent stage.

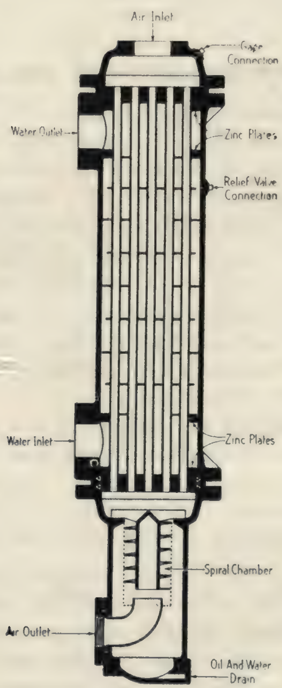
During its passage through the inter-cooler, the air surrenders to the water practically all the heat of compression, although some of this heat has already been absorbed by the water in the compressor water jacket.

After the air leaves the last stage, it is usually conducted to an after-cooler, where the heat generated in the final stage of compression is removed in exactly the same manner as in the inter-cooler.

The cooled high pressure air then passes to the receiver, where it is stored, until drawn off for use as required.

SPRAY AIR COOLER

A sectional elevation of the Schutte & Koerting Spray Air Cooler is shown in accompanying illustration. This apparatus is of highest value in successful operation of Diesel engines. The cooling water envelopes and circulates around the tubes. The compressed air enters the top header, flows through the tubes into the bottom header, thence passes into an oil and water eliminator. This is a circular chamber or tangential groove in the form of a spiral.



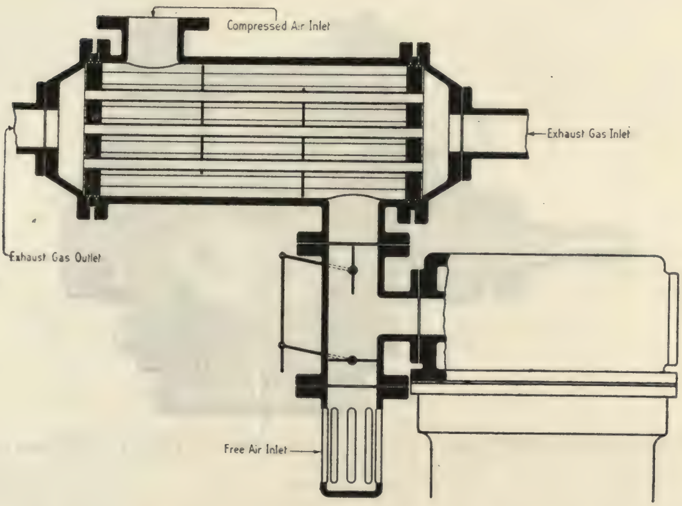
Sectional Elevation of Spray Air Cooler for Diesel Engines

As the cooled air with its entrained oil and moisture is passed through the spiral chamber, the oil and water are thrown to the outside and forced through suitable openings in the eliminator. The oil and water collect in the bottom of the cooler whence they are removed through the drain shown.

If the oil and water are not removed from the spray air, but are carried with the air into the engine, they frequently form a gritty deposit on valves, etc., which in many instances eventually causes a serious explosion.

COMPRESSED AIR PREHEATER

When heavy oils are burned in Diesel engines, it is frequently necessary to preheat the compressed air so that the final temperature of compression will be high enough to vaporize and ignite the heavy oil injected into the cylinder.



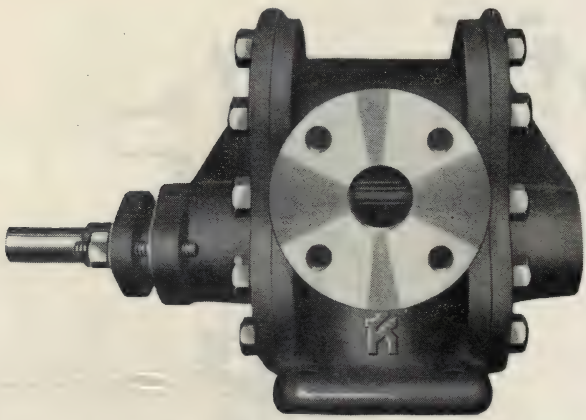
Sectional Elevation of Air Spray Preheater.

A sectional elevation of the preheater is shown in the accompanying illustration. Primarily the apparatus consists of a cylindrical tubular heater in which the exhaust gases from the engine are passed through tubes enveloped by the compressed air to be heated. The flow of heated air into the Diesel engine is regulated by means of a suitable valve.

Preheating the air decreases its density and therefore the work that can be developed by a given volume. This is of particular importance at full load. Under these conditions the amount of preheating is decreased by introducing into the compressed air a definite amount of cool free atmospheric air. The quantity admitted is closely controlled by means of a suitable valve.

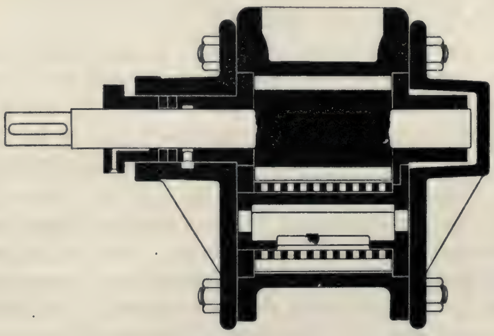
SCHUTTE & KOERTING'S LATEST IMPROVED PUMP FOR LUBRICATING OIL, THE "NEIDIG OIL PUMP"

The importance of the oil-pump on Diesel engines is best demonstrated when following the principles upon which this prime mover depends. The construction of the pump demands careful investigation of each part, corresponding with the required work to be expected of this highly important apparatus. Many defects in operation of the main unit are directly traceable to improper design and lack of sufficient knowledge on the part of builders with respect to the oil pump.



The "Niedig Oil Pump", Specially Adapted on Diesel Machinery

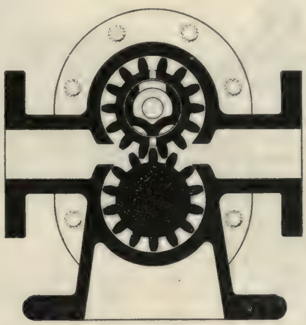
The pump used for the purpose of supplying an average stream of liquid in power plant equipments is an entirely different machine from the type used for high pressure oil-supply purpose of a high-compression Diesel engine, or for such purposes where lubricants are supplied.



Neidig Oil Pump—Interior Arrangement

In figure (a) a Schutte-Koerting "Neidig Oil Pump" illustration is given. As will be noticed from the illustrations, the pump follows along the lines of the well known gear-pump, often found in plants where heavy viscous liquids require a pump of highest grade.

A special feature of this pump is the provision of stationary guide ring, or distance ring, this is fixed concentric with the revolving gears, and, owing to the design, enables the conversion of velocity into pressure head to be very effectively accomplished, thus increasing not only the possible height of lift, but also the working efficiency of the pump from the standpoint of the desired pressure.



Neidig Oil Pump—Gear Arrangement

This pump possesses many advantages. Conspicuous amongst these being the small number of working parts, compactness, low first cost, and minimum wear and tear.

In calculations relating to these pumps the following formula will be helpful:

Let S = speed of periphery of wheel in feet per second.

Let H = height in feet to which liquid is to be delivered.

Let D = diameter of wheel in feet.

Let G = gallons of liquid delivered per minute.

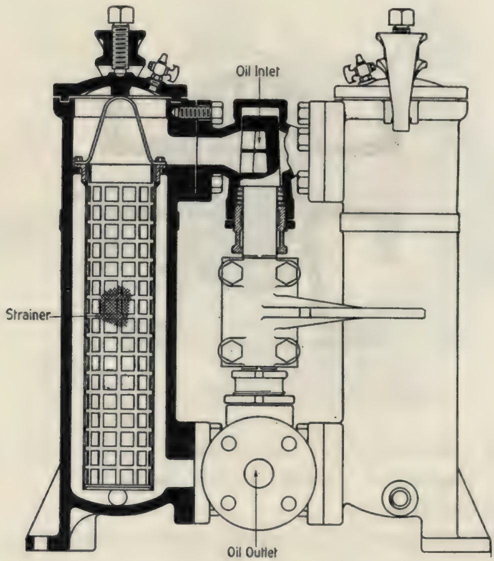
Let R = revolutions per minute.

The horsepower of driving medium required will be found by multiplying the height in feet by the quantity of liquid in pounds per minute, and by the efficiency of the pump and main unit, and dividing by 33,000. The efficiency of the pump may be anything from 0.55 to 0.65, and the efficiency of the driving power, say, 0.85, the combined efficiencies being thus equal to from 70 to 75 per cent.

Its action depends upon centrifugal principle. Until quite recently a great deal of objection was found to be prevalent to the use of centrifugal types of pumps on Diesel machinery. The principal reason of objection was a low efficiency in comparison to plunger types. These ob-

jections were never based upon sound reason, inasmuch as pumps driven by centrifugal force have many advantages, such as lessening of parts, easy maintenance, lack of complicated mechanism, etc.

In order to emphasize the necessity to provide for strainers on apparatus where the purpose is to supply oil for the engine, the sectional view of Schutte-Koerting's Duplex Oil Strainer is shown. The Duplex Oil Strainer removes dirt, sediment, and any foreign material that has accidentally gotten into the oil.



Sectional View of Duplex Oil Strainer

It is of sufficient capacity so that one side may be cut out for cleaning purposes without interrupting the flow of oil through the other side, that is, one side of the strainer can be cleansed while the apparatus is running under full power.

It is operated with a single lever. It has a free area through the straining screens that permits of long usage without causing system to become choked and clogged.

Sometimes the strainer is by-passed, the by-pass being provided with a relief valve so adjusted that it opens when a fixed differential pressure occurs across the strainer. In this way the oil flow is not interrupted; but an alarm must be provided to indicate when the valve opens, otherwise the clogged strainer will go unnoticed.

FORCE FEED OILERS

The successful operation of any internal combustion engine depends largely upon its lubrication and no matter how perfect the design, the engine will not, and cannot, run satisfactorily without particular attention to this feature. Satisfactory lubrication is more than the matter of oil. It is the question of the oil reaching the right place at the right time and the right quantity. An excessive amount of oil will cause the engine cylinders and valves to carbonize, resulting in leaky valves and loss of power. It will decrease the efficiency of the engine. The results of insufficient oil, such as burned-out bearings, scored cylinders, etc., are

"Direct Acting" Manzel Force-Feed Oiler—Fig. (1)

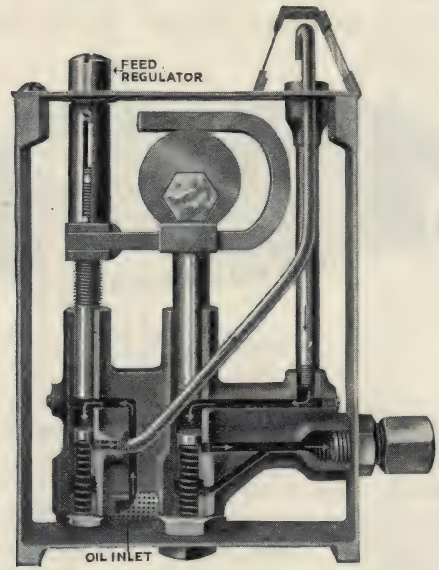
too well known to require discussion here. While in larger types of Diesels, special oil force-feed systems have been provided for, nevertheless the type made by the Manzel Co. are exceedingly satisfactory on smaller capacities of Diesel engines.

The arrangement of this type is exceedingly simple. By the use of the so-called "sight-feed" oilers, trouble will be overcome by watching the sight at the top of the lubricator. Before starting the engine, give the handle a dozen turns so as to get the oil to the bearing before the engine has started. If the engine has been left to stand, the bearing surfaces are subject to become heated, provided lubricant has not been furnished. This type of oiling system enables the operator to know—not guess—how much oil is being supplied to the cylinder bearing. The oil is always supplied in accordance to the speed of the engine, whether the engine stops, slows down, starts, the oiler corresponds to the engine's actions. It always supplies the exact amount of oil for every part of the engine, depending upon the requirements.

To regulate the Force-Feed Oiler move the stroke lever at the left in or out to shorten the stroke or lengthen it as the requirements may call for. To regulate the feed of the oil to the bearing surfaces of the cylinder, take a screw-driver and turn the feed regulator to the right or left for increase or decrease of oil to be shown at the sight. In case the oil pipes from the lubricator to the bearing surface should become clogged in any way, disconnect the pipe line at both ends, and if air is available, insert the pipe line at the end of the hose and turn on the air,

Be sure that the pipe is clean before attempting to assemble. Never let the lubricator become dry, but keep an even supply of oil in the reservoir.

Oiling by means of gravity oil cups or pressure lubricators is more or less a matter of guess work. This method is wasteful, because the feed cannot be adjusted accurately, and the oil supply is seldom proportional to the speed of the engine. Changes in temperatures affect the



Sectional View of Manzel Force-Feed Lubricator—Fig. (2)

flow of the oil, with the result that the engine gets too much or too little, and to keep it adjusted correctly requires constant attention. Even then it is not always dependable.

MECHANICAL OIL PUMPS

Where mechanical oil pumps are employed, the timing requires accurate attention. In this case it will cause the oil to be injected through a nozzle, as the piston is below the central point of its travel. The spraying of the oil takes place at this period, covering a considerable area, even though the clearance between the piston and cylinder walls is small. The piston, as it moves upward, swabs this oil over the cylinder walls.

RECORDING INSTRUMENTS

To establish an accuracy of operation in Diesel engineering, it is imperative that a plant should be equipped with instruments by which possible defects are shown. With this object in view, up-to-date plants depending upon the reliability of machinery and the engineer in charge, are placed in position to establish efficiency.

It is through the use of instruments that operators are enabled to ascertain the highest temperature in the cycle, either by cause of combustion or compression of air.

The overheating of bearings may be avoided by having instruments installed in proper places, warning the operator of approaching danger.

The storage of fuel-oils demands cautious observation, eliminating possible explosion. Instruments, showing the existing temperature should be installed.

While it is true, that engine-rooms of Diesel plants are seldom above normal heat temperature, nevertheless recording instruments should be considered a necessary equipment, in particular where Refrigeration machinery is in conjunction with the plant.

The temperature of sea water should be taken on marine work, especially on ships going on oversea voyages. This is absolutely necessary and should be recorded every 24 hours in the engine room log.

Instruments to establish the specific gravity of fuel oils should be on hand. It leaves no argument in regard to receiving the proper quality of fuel for the engine. There is very little value in water, and oils containing low specific gravity should be avoided.

Receivers of Compressors should be equipped with accurately tested gauges, Safety valves, Relief valves and Bottom blows. The latter valve is a vital equipment on compressed air-receivers. They should be placed as low as possible, to make it possible, to drain all existing water which accumulates on the bottom of the tank. Water in air is injurious to valves and piping. The acid in the water will eventually act as a detrimental factor on metal.

Pressure gauges should be tested occasionally to establish their correctness. To place reliability on mechanical contrivances, such as safety valves, relief valves, etc., should be discouraged, as it may lead to accidents of serious consequences.

The Ashton Improved Dead-Weight Pressure Gage Tester, as shown in the illustration, offers in convenient form an improved method for accurately testing pressure gauges by means of weights, and is a recognized standard extensively adopted for this important service. It is equal in accuracy to that of a mercury column, and has the advantage of being more compact, portable and much lower in cost. These testers are also much more preferred over the ordinary styles of similar designs because of their special distinctive construction with double area piston. This exclusive feature makes it possible to make tests within their designated range of pressure with only one-fourth the usual number of

weights, which is a matter of considerable convenience, as well as economy of time. In following instructions it will be observed that accuracy can be obtained when properly applied.

For low range pressure testing the tester should be adjusted so as to make use of the combined large and small area of the piston, which is done by closing the left-hand cock on the vertical pressure cylinder and opening the right-hand one. When the maximum pressure with this adjustment is obtained, and it is desired to test at higher pressures,

Ashton Improved Dead-Weight Pressure Gauge Tester

the reverse adjustment of the cylinder cocks is made with the left opened and the right one closed. This makes the machine operate on the small area of the piston only, and the pressure then exerted will be four times greater than before, which applies to the weight holder as well as to each of the weights. These changes of regulation can be made while the machine is in use and without taking it apart. It is necessary, however, to remove all pressure in the tester by unscrewing the hand-wheel before making such re-adjustments.

The tester should always be placed in a level position so that the weight piston will stand exactly vertical. To insure accuracy of readings, the piston should be revolved slowly to reduce any friction there might be in the cylinder.

As the weights force the piston to the bottom of the cylinder, the hand-wheel should be screwed in more, thus raising the piston and pre-

venting it from striking the bottom. All interior parts should be kept clean, and best results are obtained by using sperm oil or similar light grade.

In preparing the tester for use, the three-way cock on the gage connection arm should be closed by turning the lever handle to a vertical position. The hand-wheel screw should be screwed into the oil reservoir as far as it will go. Then remove cap on top of vertical cylinder and slowly fill cylinder with oil, during which operation the hand-wheel should be gradually unscrewed until the instrument is completely filled. The gage to be tested should next be applied, and the three-way cock opened by turning lever handle horizontally to the right. The weight pistons with tray may then be inserted in the cylinder, making the tester complete and ready for use with the application of the weights.

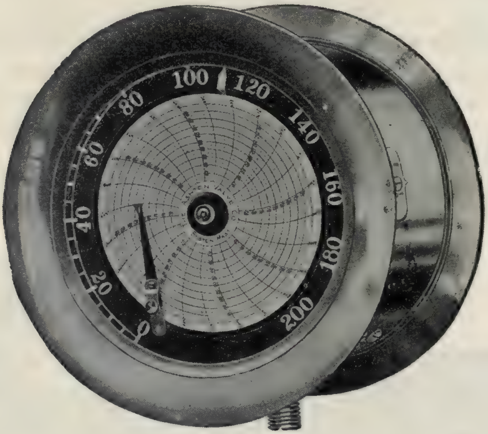
Ashton Inspector's Testing and Proving Outfit

The piston with weight holder, as well as each of the weights, is plainly marked with the pounds pressure they will exert on the gage, with double area adjustment. When the single area adjustment is being used the pressure as above stated is four times greater.

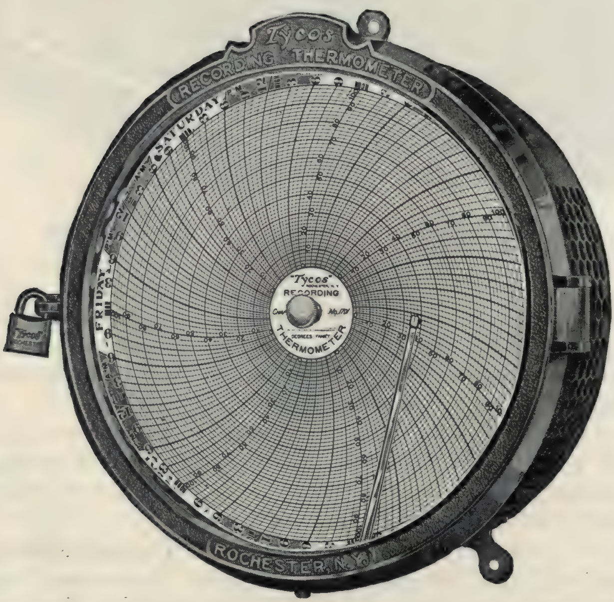
In accompanying illustration the Ashton Inspector's Testing and Proving Outfit is shown. This outfit is particularly adapted to the requirements of operators on ships going on long voyages and around plants where large Diesels are in operation.

In the illustration of the Ashton Improved Pressure Recording Gage it will be seen that the chart is graduated with pressure lines and in fractions of an hour, and is rotated by an eight-day clock movement. The chart is ordinarily made to rotate once in 24 hours.

With the use of this gage in the engine room, there is always a tendency to carefully watch the entire operation of the engine. The record of the chart shows the actual existing pressure on the air line and with the equipment of a gauge of this kind any irregularity which may cause serious breakdowns is immediately recorded.

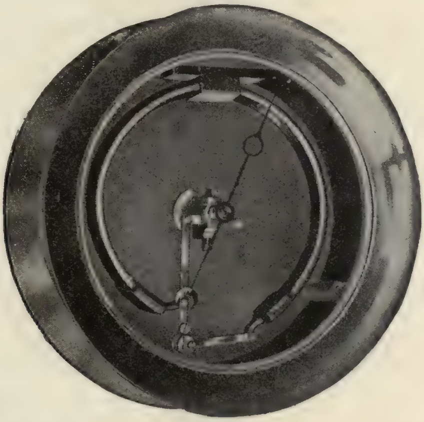


Ashton Improved Pressure Recording Gage



Tycos Recording and Index Thermometer

In the following illustration we have a Recording Instrument, which automatically writes in ink on a revolving paper chart a continuous record of the temperature to which its bulb is subjected. The self-contained recorder has the bulb, or sensitive member, inside the case, whereas with the cavillary-form instrument the bulb may be located at a distance.



Ashton Pressure Gauge—Double Spring Arrangement

An Index Thermometer is an indicating instrument having a bulb or sensitive member which may be located at a distance from the case, so that the latter can be located at a point easily accessible for reading the temperature.

It may generally be stated that the mercury type is best adapted to applications which require accurate readings over a wide temperature scale, and where the length of flexible connecting tubing does not exceed 20 or 25 feet.

Ashton Pressure Gauge—Single Spring Arrangement

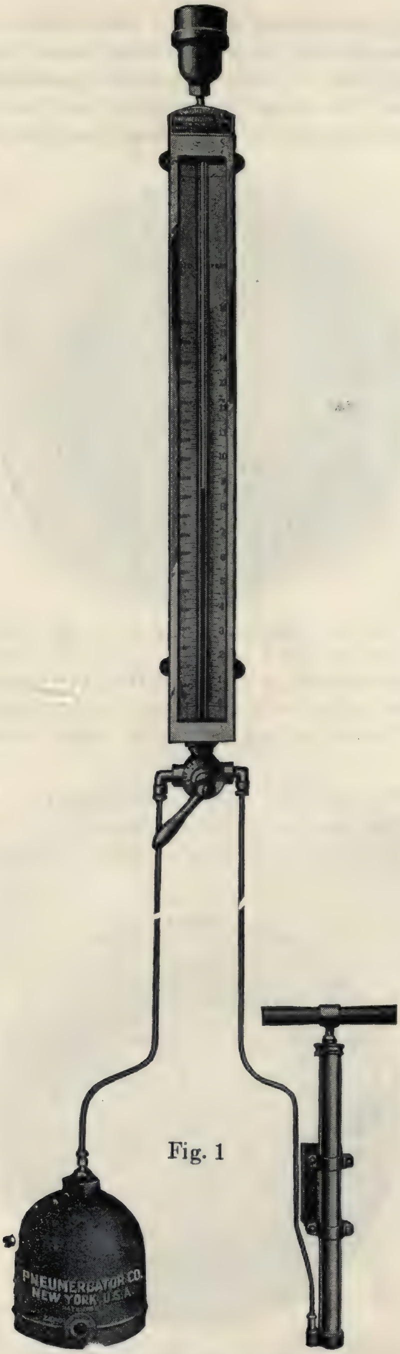


Fig. 1

Pneumercator Gauge

Instruments of this kind are made in temperature ranges within the limits of 40° below and 1000° above zero, Fahrenheit. The vapor tension type is particularly recommended in Diesel operation, where the temperatures are within the limits of 100° and 600° Fahrenheit, and particularly where long lengths of flexible connecting tubing are necessary, leading to fuel bunkers or parts of the ship or plant where a predetermined temperature is imperative.

In Figure 1, a Pneumercator Ship's Draft and Tank Gauge is shown. The application and advantages of Pneumercator Tank Gauges as applied to oil cargo, fuel oil settling, ballast or water tanks, or bilges, is recognized in marine service. They indicate the depth, volume or weight of the tank contents.

They provide an accurate and simple means of checking invoices, fillings and withdrawals, and will record the amount of fuel consumed per hour or per day. They furnish a perpetual inventory of tank contents. By their use there is no danger of overfilling or flooding decks.

It will be observed in Figure 1 that the installation is exceedingly simple. It indicates for and aft drafts of the vessel, registers mean draft and corresponding tons dead weight displacement. It weighs bulk cargoes loaded or discharged, with close accuracy, and is of invaluable assistance in trimming the vessel.

The operation of the Pneumercator Gauge is dependent solely upon the maintenance of the true static balance between the head of liquid to be measured and the column of mercury or other indicating medium in the gauge. The pressure of the liquid is transmitted to the gauge by air confined in a small connecting tube between the liquid (at the datum line above which the head is measured) and the gauge.

To establish the datum line a hemispheric vessel, or balance chamber, is located at a predetermined level below the surface of the liquid. An orifice in the lower portion of this balance chamber admits the liquid to the interior. In taking a reading, air is forced into this balance chamber, thus expelling the liquid from the balance chamber and establishing the datum level. Excess air merely passes out as bubbles, hence the pressure on the confined air remains constant and equal to the head of liquid standing above the datum line.

When, by manipulating a control valve, this air is admitted to the gauge, the mercury column rises to balance the pressure of the liquid head and establishes a precise reading on the gauge scale.

The elements required are therefore: (1) A balance chamber; (2) A mercury or other gauge; (3) A hand-pump or source of compressed air, and (5) A control valve attached to the gauge and connected by small piping to the balance chamber and to the source of compressed air. (See Fig.. 1).

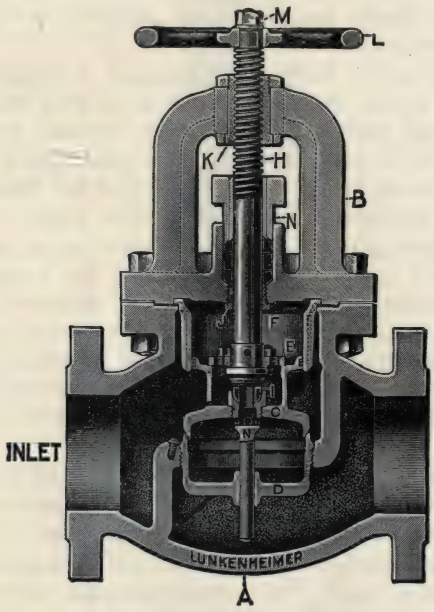
The instrument may be installed at any desired point, regardless of the location of the balance chamber. Indirect leads and any number of bends in the air line in no wise affect the working system. As the air is merely trapped in the balance chamber and piping, the pressure

which it transmits is unaffected by varying temperatures through which the latter may pass, and the instrument is of unvarying accuracy. This instrument will operate with equal accuracy on tanks open to atmosphere, or under pressure. Their precision is not affected by temperature changes.

THE IMPORTANCE OF PROPER VALVES

In selecting valves for Air-line connections or around the main or auxiliary engines, is a matter which should be given careful study. Unlike the steam engine, where a leakage of steam does not impair the operation of the plant to a great extent, the opposite may occur in Diesel operation.

In packing glands around valves, the same should be done exceedingly skillfully. Steam packing will not do around Diesel machinery, neither will steam valves be satisfactory in a plant operated by Internal Combustion engines.



Ideal Valve for Use Around Internal Combustion Machinery

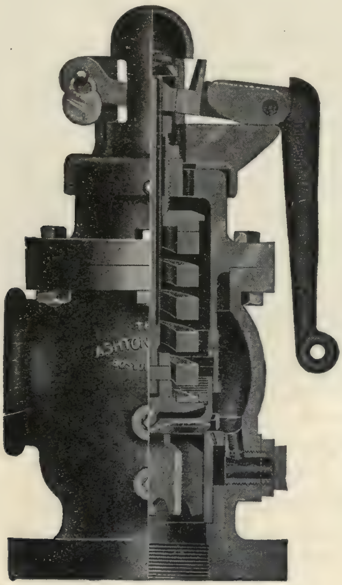
It is but natural that leakages should be avoided. It will be seen in the illustration showing the Lunkenheimer Balanced Valve, that the manufacturers have made a special type answering the purpose of proper valve equipment.

All Lunkenheimer Balanced Valves should be connected so that the inlet pressure will be above the disc. The method of operation, assuming that the valve is closed and under pressure, is as follows:

Air will pass through the drain in the disc cylinder. just above the main disc and thence through the ports E into the balancing cylinder F. The full inlet pressure will then be on top of the disc and materially assist in maintaining a tight valve.

When it is desired to open the valve, the hand-wheel L is turned about one-quarter of a revolution. The stem will then be in a position shown in the illustration with the opening to the by-pass disc I uncovered. Air will immediately pass through the by-pass disc and out ports in the main disc guide stem N, equalizing the pressure in the balancing cylinder and below the main disc.

Further turning of the hand-wheel will open the main valve, and because of its balanced condition, this operation may be accomplished with negligible effort.



Ashton's Spring Lever Pop Valve—Exposed

It will be observed that the by-pass construction of this valve not only permits the ready establishment of equalized pressure, but affords an unusually safe and accurate restriction of the volume of air transmitted to the main unit during the "warming up" process.

The small drain hole in the side of the disc cylinder also serves to relieve accumulation of water, which would otherwise leak in the

valve-arrangement on the engine when the valve is connected in a vertical position.

The disc cylinder piston ring is of sufficiently loose fit to provide adequate drainage when the valve is placed horizontally.

The removal of moisture, always present in air, is essential to safety and continuity of service. The varied requirements of piping installations make it impracticable to provide an arbitrarily located "drip" connection on the Balanced Valve, but each layout must provide adequate drainage of all points at which such accumulation would otherwise occur.

In accompanying illustration of the Outside Spring Safety Pop Valve, an excellent view is allowed, giving vital parts of this necessary equipment on Air-Tanks.

The purpose of the safety valve is to prevent the pressure of air-storage tanks from rising above a certain definite point, dependent on the construction of the tank and the condition under which it is to operate. The function must be performed automatically and under operating conditions that may arise.

There are obviously two essential requirements that must be complied with in any safety valve in order to guarantee its satisfactory performance: 1st, Mechanical Reliability, and 2nd, Adequate Relieving Capacity.

Ashton's Relief Valve

Safety valves should be connected directly to the storage tank, and in case it is found necessary that it should be connected to any outlet connection, under no circumstances should the area of such connection be less than that of the valve inlet. A close nipple should be used in case a threaded connection is necessary.

In no case should a stop valve or other fitting be placed between a Pop-Valve and the air-outlet nor on the discharge outlet between safety valve and the atmosphere.

If the laws governing the installation permit adjustment of pressure-settings, the following directions should be observed in effecting adjustment of Lunkenheimer Pop Safety Valve:

A change in the relieving pressure may readily be made by removing the cap at the top of the valve and adjusting screw—turning the latter down for higher pressure and up for a lower pressure.

The amount of pressure should carefully be determined and the setting of pre-determined pressure necessary to assure the safe carrying capacity of storage tanks or reservoirs should at all times correspond with the pressure-gauge.

SILENCERS

In the following illustration a Silencer is shown. These Silencers reduce the noise of the exhaust of an engine. In many cases these Silencers are installed both upon the inlet and exhaust, as it has fre-

Illustration of "Maxim" Silencer.

quently been found that the suction or inlet of oil engines and the section of air-compressors is the cause of nearly as much noise as the exhaust. They offer such a low back pressure that the most sensitive engines can be equipped to operate properly and quietly.

THE SPERRY MAGNETIC CLUTCH COUPLING

The Sperry type of Magnetic Coupling or Clutch has been used successfully on submersible crafts for a number of years. It may well be considered a clutch arrangement of exceptional reliability, in particular on marine machinery where power-transmission has to be depended upon assuring immediate control of the engine.

This type of clutch was developed as a by-product of several years' experimentation and development of an Electric Transmission. It has been built in large range of sizes and powers and has been found particularly adapted to certain conditions where an extreme flexibility combined with self-aligning characteristics is desired, together with speed and quick response in management of the power unit.

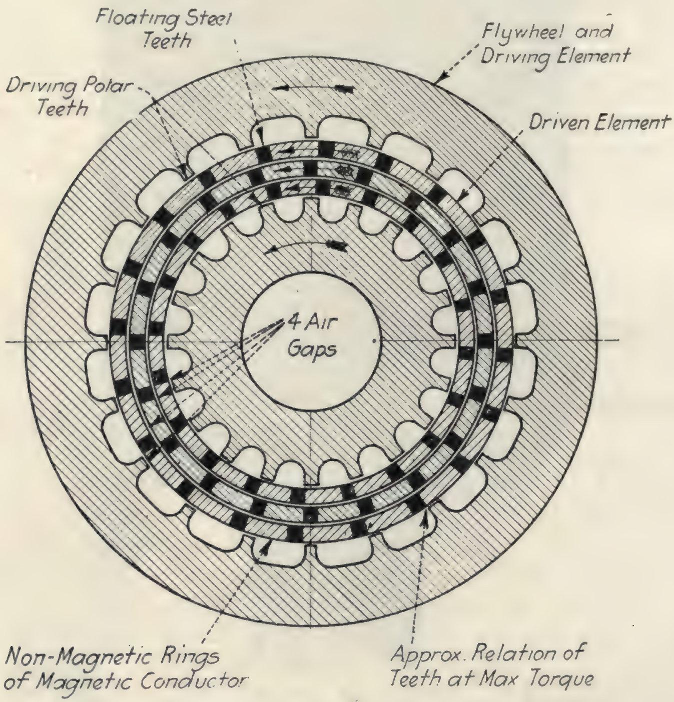


Fig. 1. Section Through Electromagnetic Clutch

In this form of Clutch Coupling torsional resistance is developed, due to the bending or distortion of the magnetic flux stream passing from the teeth of one polar projection through floating steel teeth embedded in non-magnetic material and into the teeth on the opposite polar projection. (See figure 1.) The component parts of the Coupling are so arranged as to enable the distortion of the flux stream to be made in the plane of revolution and a perfect torsional cushion is thus provided.

Simple and Rugged Construction: The component parts of the coupling are of an extremely simple form as may be noted from figure 3, all parts being substantial and rugged form. As there is no mechanical contact in the plane of revolution, the component parts are not subjected to shock of any kind and there is no liability of parts becoming loosened, worn, or broken, due to the severest condition.

The half coupling carrying the floating teeth is normally centered by a sleeve or ball bearing projecting from the opposite half of the coupling. This centering projection maintains a normal condition of the air gap and prevents exterior forces due to shaft misalignment or other conditions disturbing the proper relation of the coupling parts. The air gaps maintained in the coupling are relatively large and the floating magnetic teeth are cast solidly in a non-magnetic ring which is made by a process insuring permanent retention of the teeth and safety against all of the working and handling conditions which may be encountered in assembling and other unforeseen occasions aboard ship.

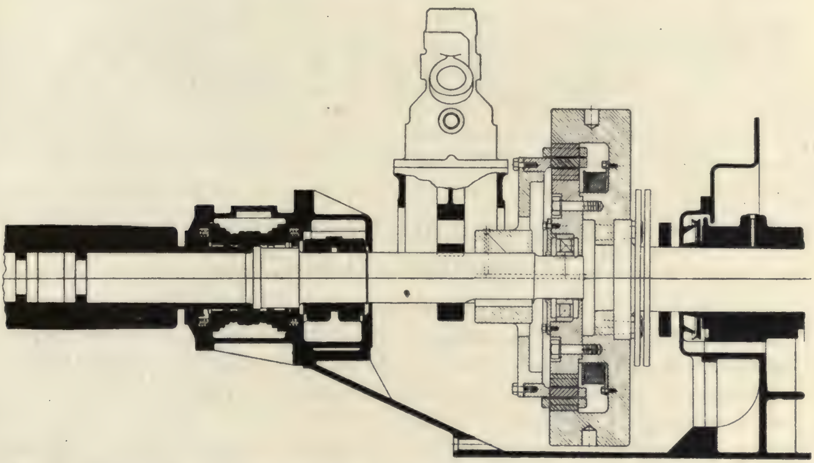


Fig. 2. General Arrangement of Sperry Gyroscope Co.'s Electromagnetic Clutch.

The energizing coil of these couplings consists of a single circular coil of relatively large size wire which is wound in a form and impregnated with Bakelite. It is insulated to withstand the worst conditions of temperature, dampness and operation. The amount of current required for energizing is extremely small for the power transmitted, as but a few watts are necessary and only a fraction of a kilowatt is required for a coupling of large capacity. The exiting current is led into the coil through a simple form of brush holder bearing on collector rings which are mounted on and form part of one of the coupling elements. The standard couplings are designed to be operated at a potential of 110 volts D. C., although exiting coils may be wound for any voltage up to 500. These

couplings are adapted only for operation on direct current circuits and will not operate on alternating current systems of any voltage.

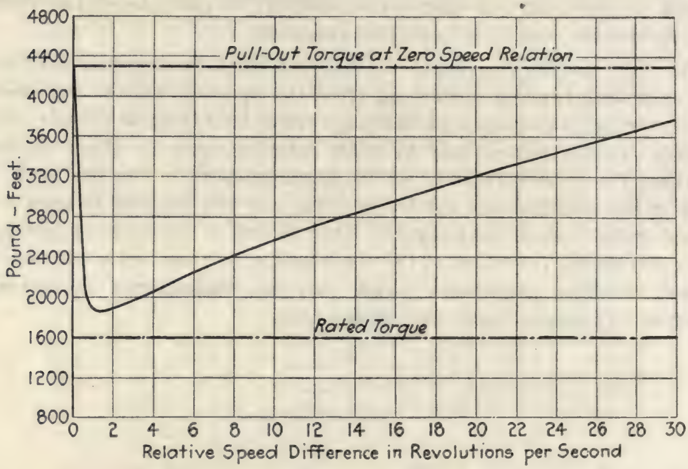


Fig. 3. Speed-Torque Curve of Electromagnetic Clutch.

TABLE OF COMMERCIAL RATINGS OF SPERRY ELECTRO MAGNETIC CLUTCH

Clutch Coupling Number *	Max. pull-out-torque lb. ft.	Normal rating constant torque lb. ft. †	H. P. per 100 rev.	Max. speed R.P.M.	Max. starting torque lb. ft. ‡
5	300	175	3.25	3000	75
9	900	500	9.0	2500	225
14	2500	1200	24	2000	625
18	4500	2200	42	1600	1250
24	10000	4500	85	1200	2500
32	20000	9000	170	900	5000
40	40000	16000	305	750	10000

*Larger sizes may be designed for special requirements.

‡Values given are for a speed difference of 1000 ft. per minute between halves of coupling. Lower speed differences give somewhat less starting torque.

†Constant torque ratings are based on such prime movers carrying smooth loads. For pulsating loads a factor should be introduced varying from 1½ to 3, depending on specific characteristics of the drive.

The clutch is energized by the simple pressing of a button and the driven part is gradually speeded up and brought into synchronism with driving part. After clutch is synchronized no slipping takes place unless a load is thrown on clutch greatly exceeding its rated capacity. This characteristic may be utilized in many applications as a guard against overloading some particular part of the system. No detrimental effect is caused by starting continuously even under load.

As will be seen in the illustration (figure 3), by carefully studying this card, that the strain in consequence of use of this type of magnetic clutch is far less than that of the different mechanical equipments generally in use on engines depending upon reverse-gear. No doubt this type of clutch has many advantages, in particular when installed on crafts where quick maneuvering must be accomplished.

REVERSE GEARS FOR MARINE ENGINES

Of vital importance is the reverse gear on marine engines depending upon this equipment. A gear must conform with the requirements expected of it. The maneuvering of the ship, and, in fact, the safety itself, depends on the reliability displayed in the reverse-gear. It must be built strong and rigid, withstanding all rough usages it is confronted with.

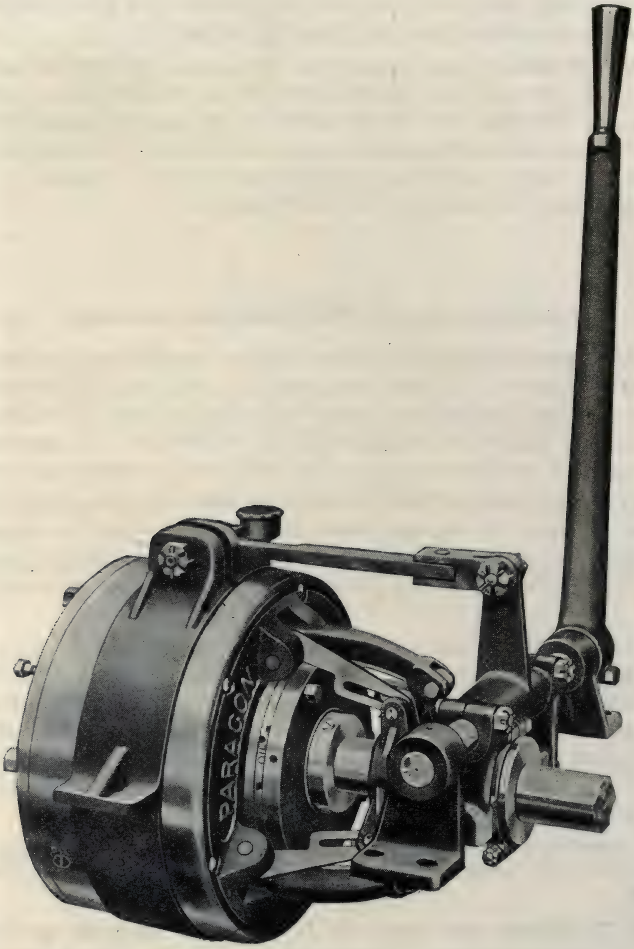
The Paragon Reverse Gear, which is shown in accompanying illustration, was designed for engines where the bed is extended to accomodate it, in conjunction with the unit power plant. It is also used in connection with any motor where a firm foundation or angle iron support is provided.

On account of its unusual compactness, it takes up a very small amount of room. The forward end of the gear is bored out to directly accommodate the crank shaft of the motor. The propeller shaft can be fitted directly into the rear end of the gear, which is bored out to the propeller shaft size. Ingenious stop links lock the gears securely in either position. This type permits the operating lever to be placed on either port or starboard side.

In following detailed explanation pertaining to the adjusting of the gear, by carefully using the illustration shown here it will be found instructive.

How To Adjust the Gear: It is necessary that the gear should be properly adjusted before it is permitted to operate. The forward drive of the Paragon gear is obtained by means of the disc clutch which locks to the case. Thus the whole gear revolves as a solid coupling. The locking, or clamping, of these discs is brought about by the pressure produced by the leverage obtained through the combination of the operating lever and the expansion of the fingers. Unless this pressure on the discs is great enough, the clutch will slip and heat. Consequently, the discs will be cut up and their carrying power destroyed, thus necessitating the purchase of new parts.

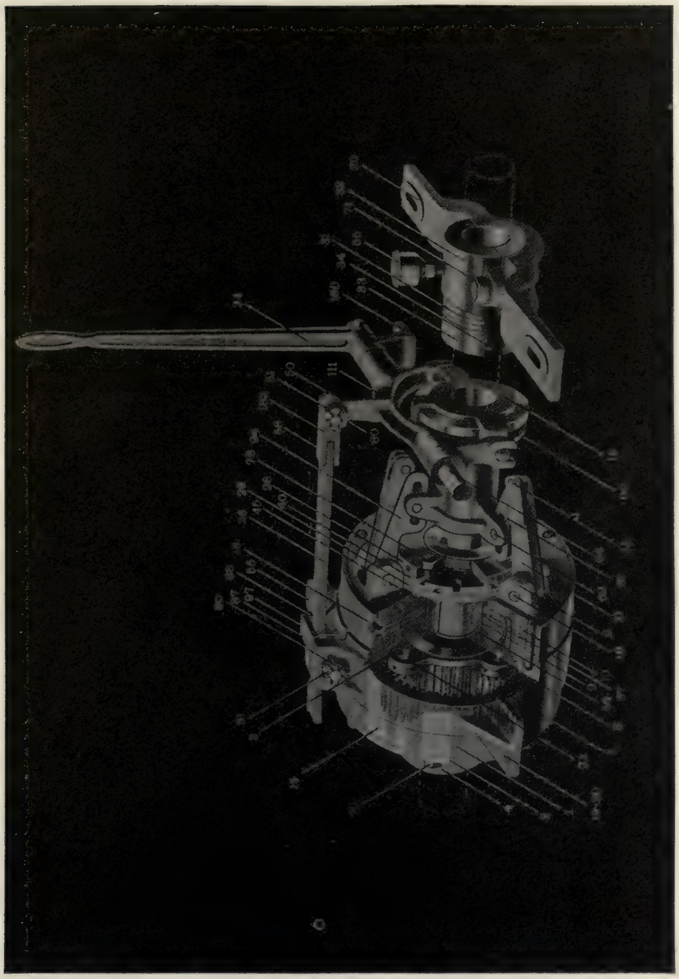
Remember, that every time the clutch slips the plates become thinner and further adjustment is necessary to take up the wear. It is, therefore, necessary that this adjustment should be obtained before the clutch is allowed to run at all.



Yoke Operating Type of Paragon Reverse Gears

If the gear heats on the forward drive it indicates the gear is slipping and should be adjusted at once.

If the gear slips on the forward drive, back the set screw (76) out of its notch in the brass check collar (40). Turn the screw collar (28) to the right until the set screw (76) projects into the next slot in this brass check collar (40). Then tighten the set screw.



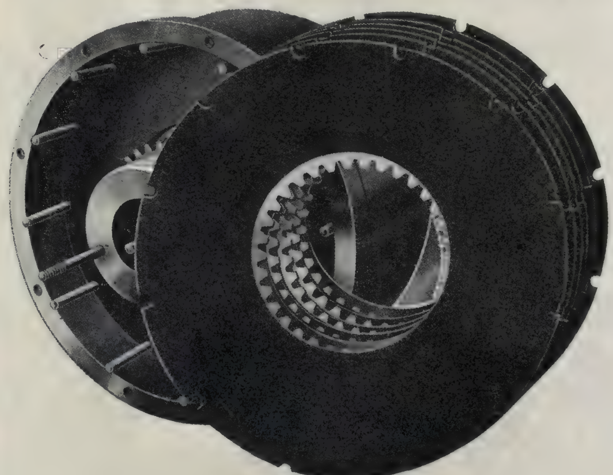
Phantom View of Paragon Reverse Gear, Showing Parts

If the gear still slips, back out the set screw again and turn the screw collar (28) to the right so that the set screw projects into another notch. If it still slips, repeat the process until the gear does not slip.

In case, however, the adjustment is too tight after taking it up one of these notches, an arrangement is made for taking up this adjustment a half notch. To do this, back the set screw completely out of the hole in which it is placed and insert it in the other hole on opposite side of the screw collar (28). Then turn the collar to the right until it projects into the next notch on the brass check collar (40).

In all cases be sure the set screw goes back into one of the notches on the brass check collar.

The neutral position is obtained when the operating lever is vertical. The reverse position is obtained by means of the brake band which clamps around the case and keeps it from revolving. The brake band is operated by throwing the lever back as far as possible.



Friction Assembly for Forward Drive

If the gear slips on the reverse, that is, if the case (1) revolves when the lever is in reverse position, make adjustment as follows, while the motor is running slowly; remove the cotter pin from the nut (51) at the top of the brake band and tighten this nut until the case ceases to revolve, keeping the lever thrown back as far as possible. When this is done replace the cotter pin.

Directions for Lubrication: In some motors the matter of reverse gear lubrication is taken care of automatically by the same lubricating

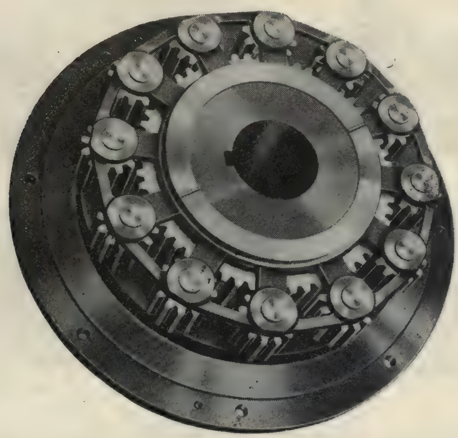
system which lubricates the motor. In such cases, hand lubrication is, of course, unnecessary.

When the reverse gear is placed in an oil tight compartment, and a splash system of lubrication is in use, be sure that this compartment is kept at least half full of a good grade of lubricating oil.

If neither of the above lubricating systems are used in connection with the motor, it will be necessary to lubricate the gear by hand in accordance with following instruction:

Before running the gear remove the brass plug (15) in the case and put in non-fluid oil. This is really a grease instead of an oil and is about the same consistency, or thickness, as vaseline. In gears of larger sizes brass plug (15) will be found on front cover.

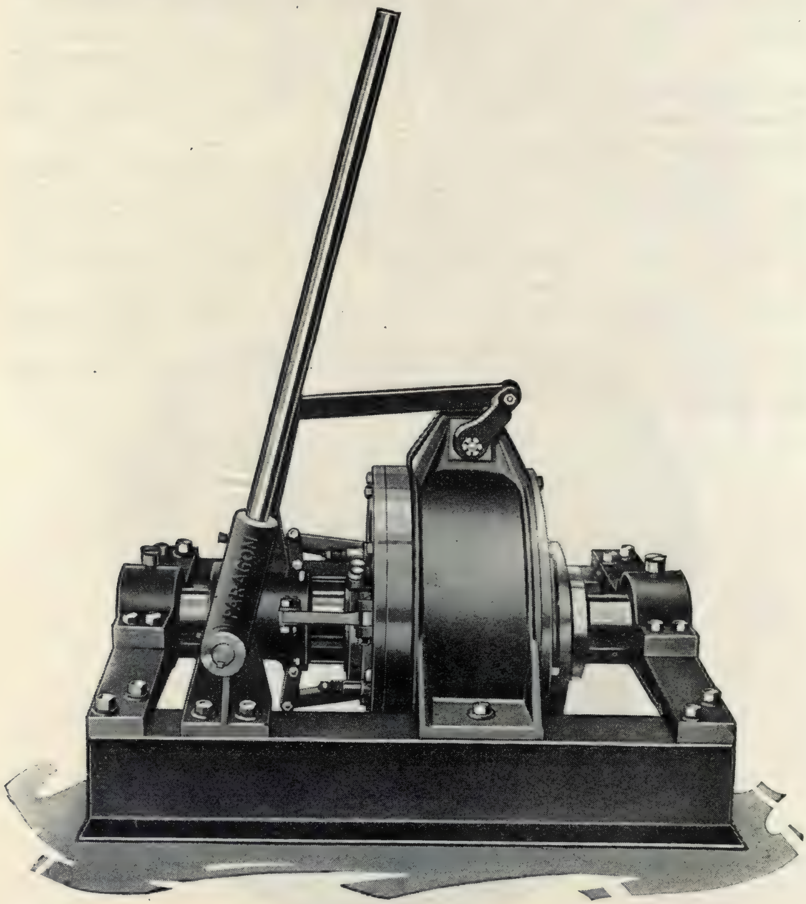
After this has been done, remove the brass screw which is located in the case on the other side of the brake band. For lubrication here pour in the equivalent of two or three teaspoonfuls of cylinder oil which is for lubricating the discs. While this is being done the lever should be in reverse position so that the plates will be freed from each other. Keep turning the engine over by hand, or run the engine slowly while injecting this oil. Oil the brass collar and the disc (40) at the place stenciled "oil". Keep all grease cups filled and screw them down as frequenty as necessity requires it.



Gear Assembly for Reverse Motion

The tremendous explosion impulses of a slow turning oil motor demands a gear of unusual holding power. In following illustrations types of gears are shown adaptable for Diesel-powered marine engines.

In the clutch assembly of Model "H" illustrated here, consisting of five cast-iron friction discs which are ground for smoothness. The internal teeth of these discs mesh with the external teeth of the propeller gear hub as shown here. There are also six bronze friction discs which are held in place to the case by twelve studs. These studs are distributed along the circumferences of these discs and are supported at both ends.

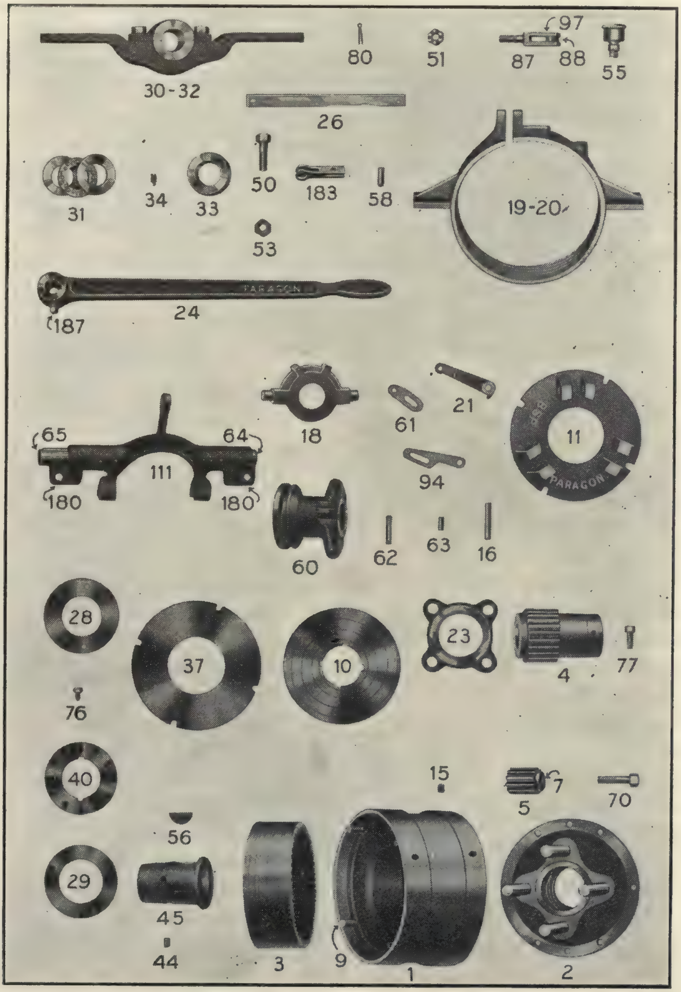


Extra Heavy Duty Type of Paragon Reverse Gear

These eleven friction discs and their adjacent surfaces, comprising a total of 24 friction surfaces, furnish a total friction area of over 2500 square inches. This is one of the reasons why this type of Paragon has met with such a pronounced success in connection with powerful oil-burning motors.

As may be seen from the illustrations, the engine gear of this model is built specially large and strong. The power is transmitted from the engine gear to the propeller gear through a single train of twelve pinion gears, thus distributing the load over an unusually large number of intermediary pinions all in one place.

The operating mechanism is of the double incline lever type and permits of the operating lever being placed on either the port or the starboard side.



Itemized Parts of Paragon Reverse Gear

In maneuvering of engine it is imperative that reversing should be accomplished in the quickest time possible. Power transmission depending upon the strong pulling capacity and reliability in service, calls for strong and rigid built gears.

The motor's power in Paragon gears is transmitted from the engine gear direct to the propeller gear through four pinions, each equi-distant from the center and distributing the load evenly. This makes a direct short line for the power to travel.

In some motors the matter of reverse gear lubrication is taken care of automatically by the same lubricating system which lubricates the motor. In such cases, hand lubrication is, of course, unnecessary.

When the reverse gear is placed in an oil tight compartment, and a splash system of lubrication is in use, be sure that this compartment is kept at least half full of good grade of lubricating oil.

If neither of the above lubricating systems are used in connection with your motor, it will be necessary to lubricate gear by hand in accordance with following instruction: Before running your gear remove the brass plug in the case and put in non-fluid oil. This is really a grease instead of an oil and is of about the same consistency, or thickness, as vaseline. The brass plug will be found on the front cover.

DEFINITION OF PARTS OF PARAGON REVERSE GEAR

1. Case assembled with Friction Disc Pins No. 9 and Oil Plugs No. 15.
2. Cover assembled with Cover Bushing No. 6 and Pinion Studs No. 8.
3. Propeller Gear.
45. Propeller Gear Hub.
4. Engine Gear.
5. Pinion Gear assembled with Pinion Bushing No. 7.
10. Friction Disc.
11. Finger Disc.
21. Fingers.
61. Toggle Link.
16. Finger Pin.
63. Toggle Link Pin.
94. Cone Stops.
18. Toggle Collar.
19-20. Brake Band.
23. Pinion Support.
24. Lever.
187. Lever Set Screw.
56. Woodruff Key for Lever.
26. Locking Link.
28-76. Screw Collar.
76. Screw Collar Cap Screw.
29. Finger Collar.
37. Friction Disc.
40. Check Collar.
44. Check Collar Dogs.
50. Locking Link Bracket Screw.
51. Castellated Nut.
53. Hexagon Nut.
55. Grease Cup.
60. Toggle Sleeve.
62. Toggle Sleeve Pin.
70. Cover Cap Screw.
77. Engine Gear Set Screw.
80. Spring Cotter.
87. Adjusting Belt.
111. Yoke assembled.
180. Yoke Shaft Brackets.
30-32. Rear Bearing.
31. Thrust Bearing.
33. Thrust Collar.
34. Thrust Collar Set Screw.

In following illustrations, showing detailed parts of the Johnson-Carlyle type of friction clutch, a clear idea will be obtained as to the distinctive feature of this reverse gear. The illustration, figure a., shows in perspective a nest of spur gearing, incorporated within a clutch body or gear cage, on each end of which are mounted clutch members of the Johnson type of clutch. These gears run on four hardened shafts, each end of which are supported in the ends of the gear cage. The gears are always in mesh with the engine and propeller shaft pinions, as shown, and the former extends to the right and the latter to the left, each being supported in babbitt bearings in the ends of the gear cage and extending through far enough to be coupled onto.

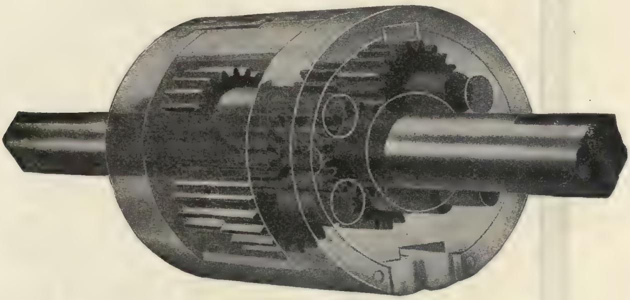


Fig. (a). Double Clutch Gear Cage Perspective

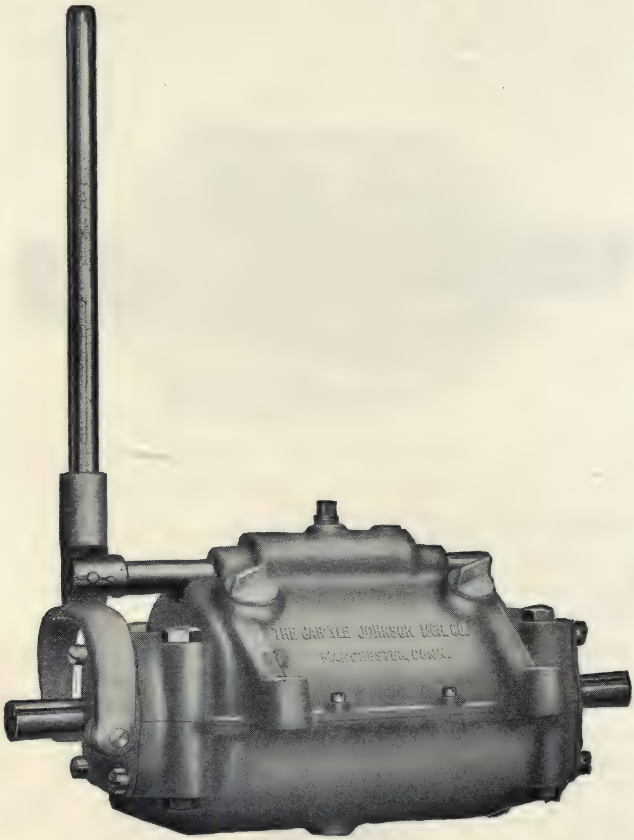
The gearing and shafting are small in diameter, in order to keep the construction compact, but are made of alloy steel, heat treated and hardened, thus giving these parts the strength of a cast-iron or machine steel gear several times as large.

The expanding friction rings, figure b., are shown one on each of the gear cage, with a set of toggle levers in each, diametrically opposite, for use in expanding same. In the friction cups in which these rings expand, surrounding the rings are placed in such position to take up all leverage required of the same.

Fig. (b). Double Clutch, Gear Cage

Spaced midway on the clutch body is the shipper sleeve with two hardened curve-shaped wedges riveted in it. These wedges force the levers apart, thus expanding the rings, bringing their outer surfaces into frictional contact with the inner surface of the friction cups.

The leverage is so compounded that it requires but little pressure to operate the clutches. The adjustment is very simple, as one screw which moves two taper blocks set into the base of the toggle levers adjusts the contact of each ring and cup to any tension. This screw is easily reached with a screw driver through a hole in the reverse gear cover and friction cup.



Propeller End *Fig.* (c). *Exterior View* *Engine End*

In the double-clutch construction the hub on the friction cup is clamped on its outside diameter within the cases and contains a combined radial and double thrust ball bearing, through which the propeller shaft runs. The hub of the other friction cup is free to revolve in the casing, within a radial ball bearing, while the engine shaft of the gear extends through it and is keyed therein.

Adjustment: The forward drive clutch can be adjusted independent of the reverse drive clutch. If either forward or reverse clutch shows any tendency to slip, it should be adjusted at once.

As there are only two points of adjustment in this gear, the operator will have the minimum amount of trouble if the clutches are kept adjusted to a tension where they will not slip, provided the gear is being used for power within its rating.

To adjust the engine and clutch, remove the thumb nut nearest to the engine, on the top of the reverse gear case, turn the engine and shaft until the hole in the friction cup comes into view, then turn propeller end shaft until adjusting screw appears under hole in friction cup. With a screwdriver with a fine point turn the screw a fraction of a turn, or more if necessary, to the right to tighten, to the left to loosen. To adjust the propeller end clutch, remove the other thumb nut, and turn the propeller shaft until the adjusting screw appears through hole in friction cup, this latter being stationary on this end of gear. Adjust as above.

Lubrication: The lubrication automatically takes care of itself if the gear case is supplied several times during the season. (Fill about one-half full).

Do not run the gear without sufficient lubricant in the case. A gravity system of lubrication leads to all bearings in the gear.

To fill the case, remove the two thumb nuts on the top cover and oil or grease can be put in at these points. Do not use a hard, heavy grease, as the design of these gears is such as to require a medium heavy oil or grease. The oil shedders on the inside of each end of the gear case prevent the lubricant from working out.

ELECTRICAL AUXILIARIES

As a result of the use of electric motors on Diesel engine-driven vessels and their longer and extensive use in the navy, the marked advantages of motor-driven auxiliaries are now recognized as never before.

The adoption of electric drive for auxiliary machinery is bound to increase on vessels propelled by internal combustion engines, in particular larger types intended for long voyages. In the many large Diesel-propelled ships, where electric steering gears have been in use as well as winches, it has proven an exceeding reliable and above all economizing factor. There are numerous other advantages in using electrical equipment in conjunction with Diesel power, which we will undertake here to summarize as a few worthy of mention:

(1). Comparative little expenses in maintainance.

(2). More reliable speed control is obtained.

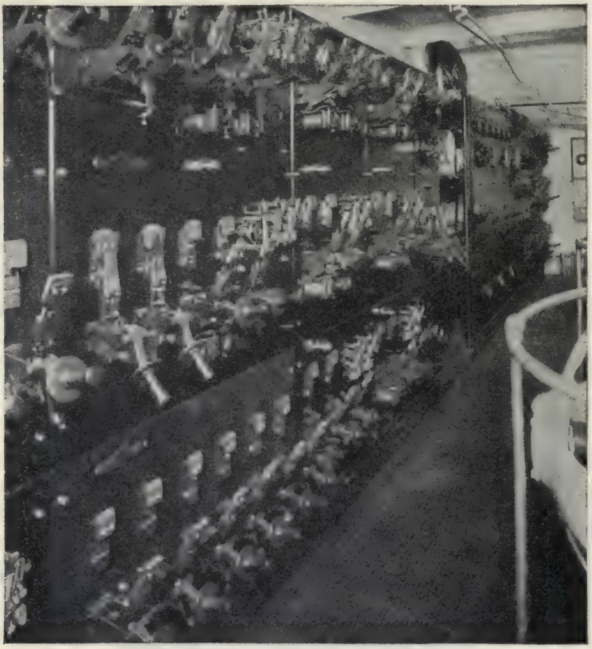
(3). Better methods of control.

(4). Electric power consumption can be accurately measured.

(5). Electric power use is cleaner and quieter than other powers.

Reasons given here could be followed up with numerous others, but they are sufficiently convincing to show to the most skeptic that with the introduction of electrical units a great many advantages may be obtained. Motors require much less labor expense than engines. One man can usually look after many motors, but cannot properly handle more than one engine under the same conditions.

Modern marine electrical equipment is designed for the most part, for 230-volt service. Several years ago the U. S. Navy Department adopted this voltage for practically all capital ships. The use of this higher voltage has decreased the size of generating equipment, motors, and wiring, as compared with that which was required with the low voltage systems used in the early days. The higher voltage has, at the same time, been found just as satisfactory from an operating standpoint.

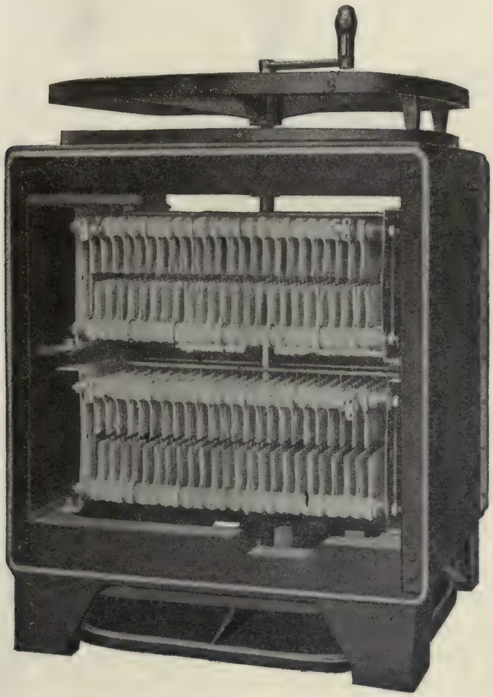


A view of the engine room switch board on the motorship, "Solitaire," showing the circuit breakers and several of the C-H Magnetic Contactors, by means of which the engine room auxiliaries are controlled.

Marine control equipment for motor-driven auxiliaries must be adapted for the conditions found on board ship in order to insure satisfactory operation. As a rule, the characteristics of marine controllers differ from similar equipment used on land. Early failures of electric drive in marine service were traceable to equipment that was manufactured primarily for use on land or that was designed by those who were

unfamiliar with the actual requirements of marine service. Illustrations shown here are products of the Cutler-Hammer Company of Milwaukee, which firm has added many late features commendable for Diesel-powered ships where electrical equipment is called for.

Marine control equipment should, generally speaking, be constructed along more rugged lines and should have a larger factor of safety than is necessary for similar equipment used in other industries. Repairs are not as easily made in the marine service as on land. Consequently, marine controllers must be capable of withstanding rough handling by inexperienced men with a minimum of maintainance expense.

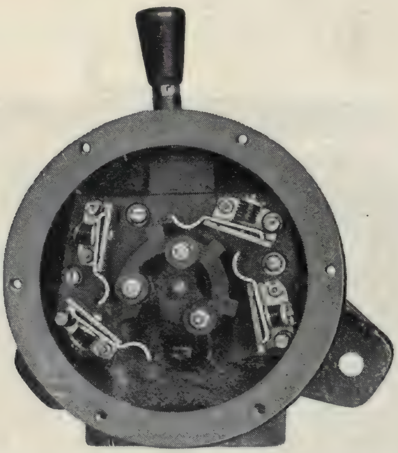


C-H Water-tight Rheostat (same as shown at left) with upper and lower covers open for ventilation. The front cover also has been removed to show the resistor units. This cover is not removed after installation except for occasional inspection.

All marine controllers should be made as simple as possible to facilitate maintainance work and the prompt location of trouble. All parts must be carefully protected from salt air and moisture conditions, and controllers used on deck or where subject to possible water pressure must be made water-tight. For certain navy uses and for use on the merchant marine where inflammable liquids or gases may be present, gas-tight con-

trollers are made specially for this service. Controllers used with certain auxiliaries are so constructed that unauthorized manipulations are impossible.

Automatic controllers used on board ship are provided with contactors which cannot close accidentally due to the rolling of the ship or open because of violent shock. All contactors must usually operate satisfactorily when inclined at an angle of 30 degrees from the vertical in any direction. All contactors used in marine service are subject to more or less vibration and should be protected from the loosening of parts by suitable locking pins or nuts.



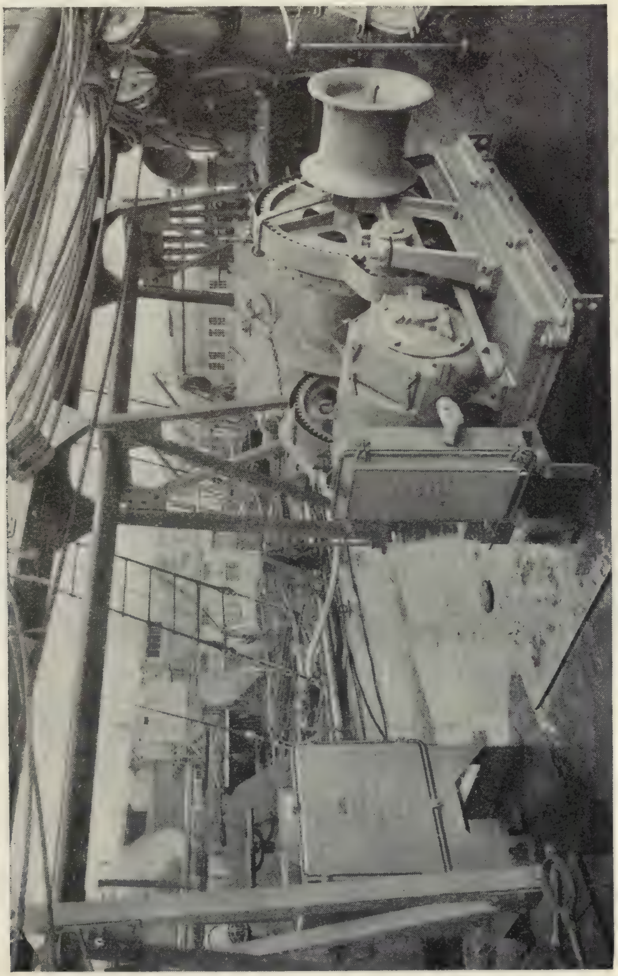
C-H Master Switch of the type used for operating small steering gear controllers like the one shown above. Cover removed.

Resisters used with marine controllers must be carefully protected from corrosion and from damage by shock or vibration. Some resistors are used where the ventilation is restricted and in such cases correct design is very essential. Resistors used on deck where they may be subjected to immersion are protected by water-tight enclosures so arranged that they may be opened for ventilation when in actual operation. Resistors used in marine service are usually built up in such a way that single resistance units may be replaced readily, if damaged.

Other requirements, which are peculiar to marine control equipment and which are necessary to the successful operation of motor-driven auxiliaries, might be mentioned. Those already outlined in previous pages will serve to indicate the necessity for studying the actual conditions on board ship before designing marine controllers and such apparatus imperative in general operation.

Illustrations shown here of the motorship "Solitaire", a steel tanker of 6730 tons displacement, launched in 1920 at Bath, Maine, is an excellent example of the new American merchant marine. The auxiliaries are

all motor-driven. Engine room auxiliaries are controlled from a centralized point. The electrician on watch is responsible for starting the various pumps, etc., when signalled by bell and pilot light from the unit to be started. The signal is given by pressing a push button at the auxiliary; it continues until the electrician closes the line circuit breakers. The auxiliary motor may be stopped automatically by pushing the stop button located nearby.



Motor-operated deck winch on the motorship, "Solitaire," equipped with a C-H Watertight Drum Controller and Resistor. The drum is located where the operator has a clear view of the winch and its load and where he can operate the foot brake.

After the line circuit breakers are closed, the motor is accelerated automatically through magnetic lockout contactors which quickly bring the auxiliary up to speed. The motors are thus protected from injury due to improper handling and manipulation by inexperienced operators.

The deck auxiliaries on the motorship "Solitaire" are handled through manually-operated controllers from stations convenient to auxiliaries. The capstan motor is controlled by means of a drum controller located on a bulkhead below deck. The drum controller is fitted with a shaft which extends up through the deck and on the top of which is an operating handle. This handle is located adjacent to the capstan barrel so that the operator has an unobstructed view of what is going on. The deck winch is controlled by a water-tight drum controller located where the operator has a clear view of the winch.

C-H Automatic Steering Gear Controller of the type used on merchant vessels. This controller is installed below deck and is operated from a master switch similar to the one illustrated below.

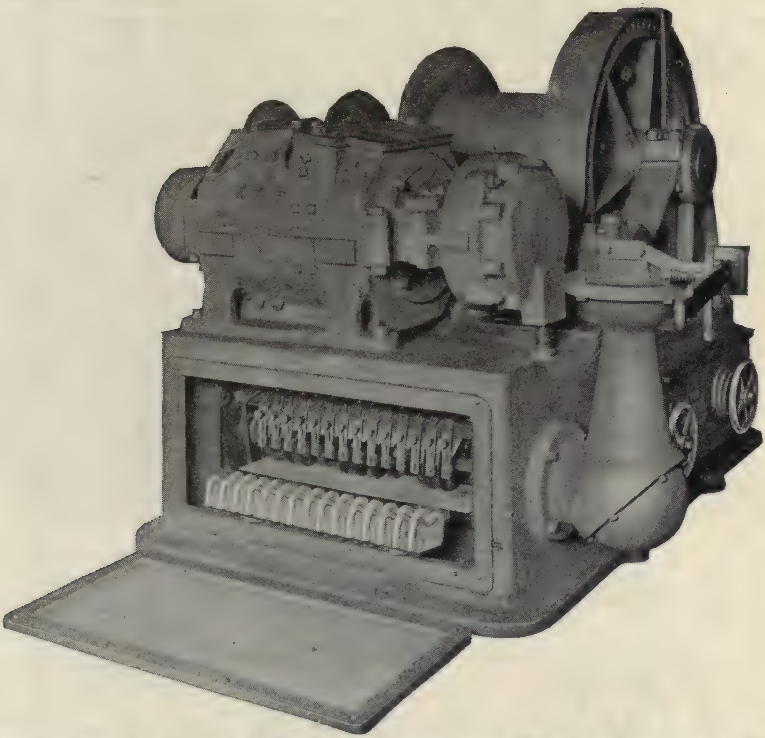
In addition to the electrical control equipment found on the "Solitaire", electric heaters are used for heating the quarters of the officers and crew. Electric heating eliminates all other provisions necessary to heat the living quarters of the ship in colder climates. Three heats—low, medium and high—are provided, thus insuring comfortable quarters under all weather conditions.

Electric current, being easily transmitted, is not subjected to losses as in the case where Diesel-powered ships have steam auxiliaries in conjunction and through condensation and latent losses extravagance is bound to occur. A saving in space and weight is made; greater flexibility and ease of control are effected. Where once the operating engineers have sufficient training in electrical matters (which is a requirement not to

be overlooked) repairs, adjustments and maintainance, can be cared for more easily than similar work in connection with other powers.

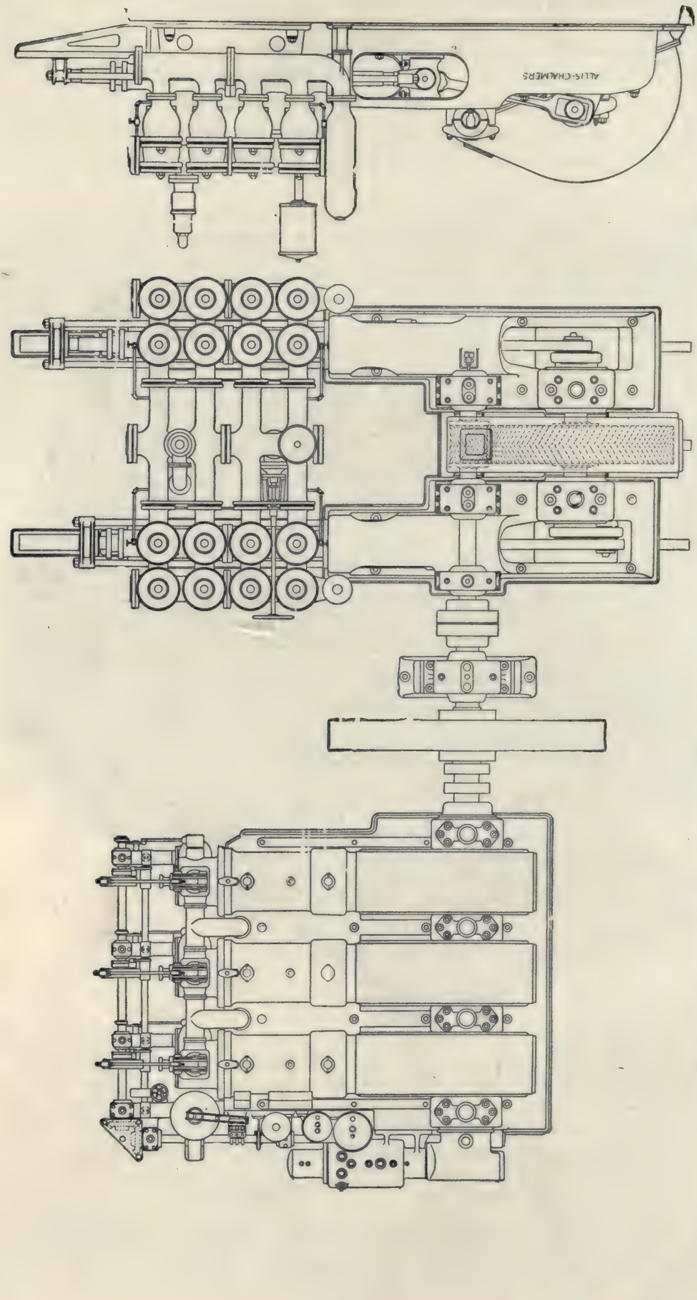
It is a well known fact to all familiar with the running of ships, that sometimes more care is necessary and more trouble caused in keeping the auxiliary machinery than the main engines of a vessel in working condition. The consumption of power for running the auxiliaries is also very large because, as a rule, these auxiliaries are driven by direct-connected engines of a simple and poor construction working at full admission; besides, a considerable loss of power is due to the extensive net of piping.

Of late, endeavors have been made to remedy this trouble by introducing in first-class modern vessels, winches and steering gear worked by electricity.



Motor-driven Cargo Winch equipped with a C-H Drum Controller. The drum and resistor are mounted in the water-tight base, the front cover of which has been removed to show the drum. The arc shields have also been swung back. The resistor is installed in the far end of the base where hand wheels, shown on the lower right hand side of the illustration, permit the opening of base covers for ventilation while the motor is operating.

The designing of all auxiliary machinery of Diesel vessels for electrical power is very desirable and this method is being brought in practice by many builders of Diesel machinery.



Plan View of Allis-Chalmer Diesel Engine. Excellent Power Plants for Electric Generation

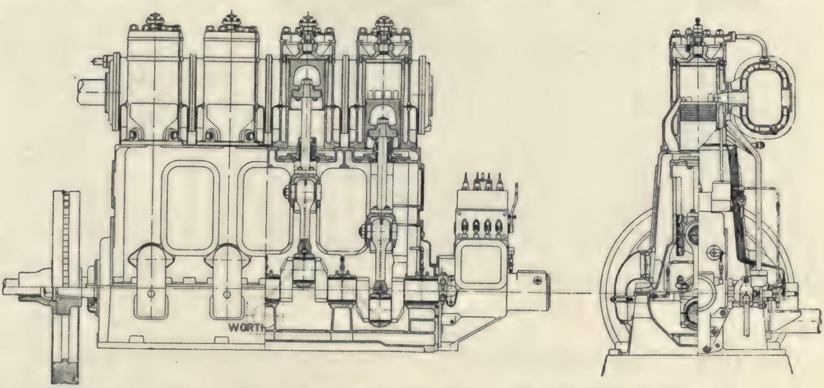
The power is supplied by auxiliary Diesel engines, installed along the side of the engine room and directly coupled to continuous dynamos from which the current is conducted to a switchboard and thence to all the auxiliaries of the vessel. The auxiliary Diesel engine system is carried out in quite a special manner dimensioned to suit the requirements on board, so that under all conditions full certainty is attained of having at any time at disposal the necessary current, which affords the absolute reliability required. The fact is that the absolute certainty of current always being at hand is a condition necessary for the system operating in a perfectly satisfactory manner, seeing that the working of an auxiliary of such importance as the steering gear depends thereon. The auxiliary engines are thus dimensioned that they are partly able to generate the current required for working all the loading and discharging winches in port, and partly for generating the smaller amount of current required for keeping the auxiliary engines of the vessel running.

The main engines of the vessel are self-contained, the pumps being worked independently.

Electrical driven machinery run by dynamos and worked by auxiliary Diesel engines affords a high degree of economy, as the consumption of fuel oil for running all the auxiliaries of the vessel at sea amounts only to a few per cent of the consumption of fuel oil for running the main engines, and the electrical power required for running the loading and discharging winches in port is performed with a consumption of only one-tenth part of that necessary for steam-driven winches.

The electric cables are easily and suitably laid from one end of the vessel to the other and do not require the space or attendance required by the pipes.

On land electric transmission of power has, by degrees, superseded all other systems of transmission; it is therefore quite natural that also on board ship electric transmission will displace all other systems of transmission, the electric machinery must of course be designed specially



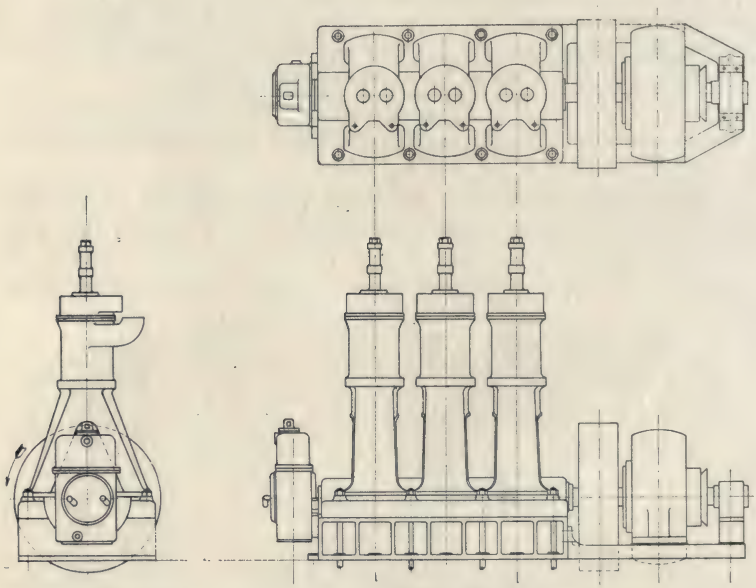
Diagramatical View of Worthington Diesel Engines, Suitable for Auxiliary Purposes

for marine purpose, heavy and strong, with motors, controllers, etc., of the water-tight enclosed type; the motors supplied for this purpose must be of the best material suitable for this service.

Steam-driven and distillate-driven auxiliaries have been found to have certain disadvantages on board the full-powered Diesel mortorships. The boilers used in connection with the steam-driven auxiliaries have given no end of trouble and annoyance, and when small distillate engines are used, the necessity of carrying two kinds of fuel (crude oil for the main engines and distillate for the auxiliaries) when storage space is at a premium, is a decided drawback.

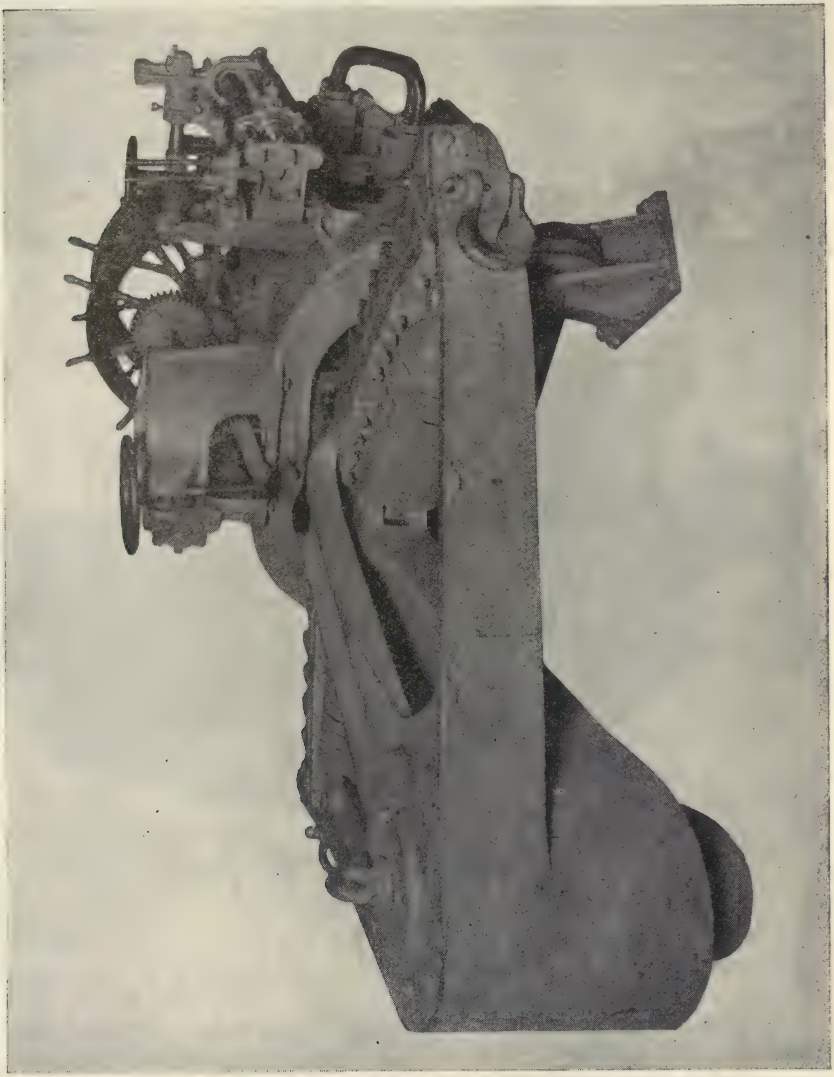
In accompanying illustration, the diagram shows a general arrangement of system adopted by the Dow Diesel Engine Company in the plants constructed by this firm. These specially constructed Diesel engines are direct-connected to electric generators, which in turn supply the power for all loading and unloading purposes, light, etc.

As will be seen in this diagram, the Dow Full Diesel Type Crude Oil Engines direct-connected to electric generator is a compact, self-contained unit.



Plan View of Dow Diesel Engines, Direct Connected to Electric Motors

The accompanying illustration shows a Allan-Cunningham Hydraulic Electric Steering Gear. This machine has been specially developed for use on marine service of Diesel-powered ships. The machine is equipped with hydraulic telemotor control, directly operated from the Wheel House.



Allan-Cunningham's Hydraulic-Electric Steering-Gear. This Type Has Proven a Reliable Machine for Diesel-powered Ships

In this machine a constant speed, continuously running electric motor furnishes the power to operate a spring quadrant mounted on the rudder stock, and this is applied through a hydraulic variable speed gear, whose speed and direction are controlled by means of the hydraulic telemotor

system from the wheel house. Follow up gear is provided so that the rudder movement follows closely the direction and amount of movement of the wheel by the man steering. Its reliability is primarily due to the accurate close mesh gears which quickly correspond to the movements of the ship.

The Allan-Cunningham type of electric Anchor-Windlass is shown in the accompanying illustration. The anchor windlass is a piece of equipment that is not often put to use on board ship, but when it is called upon for service, needs to perform some very severe duties for a short time, or withstand some very heavy labor. For this reason ruggedness and strength of all the parts should be its principal characteristic, as shown in the illustration, where a marine type series motor is geared through spur gears, worm and worm gear to an intermediate shaft on which are mounted the gypsy heads for use in warping ship. The intermediate shaft carries two sliding pinions for meshing with the main gears on each

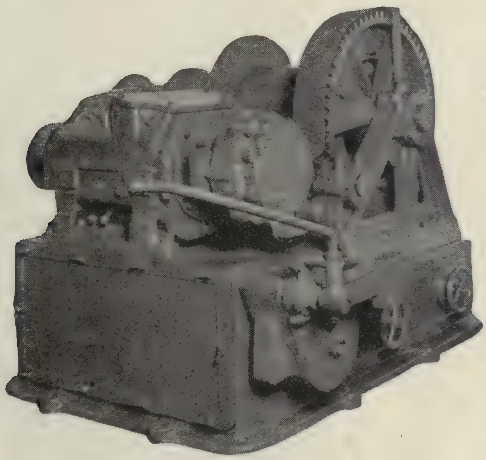
Allan-Cunningham Electrically Operated Anchor-Windlass—Specially Constructed for Diesel-powered Ships

wildcat. A clutch-shifting device provides for throwing the wildcats in or out of gear, and band brakes can be used to hold them in any position desired.

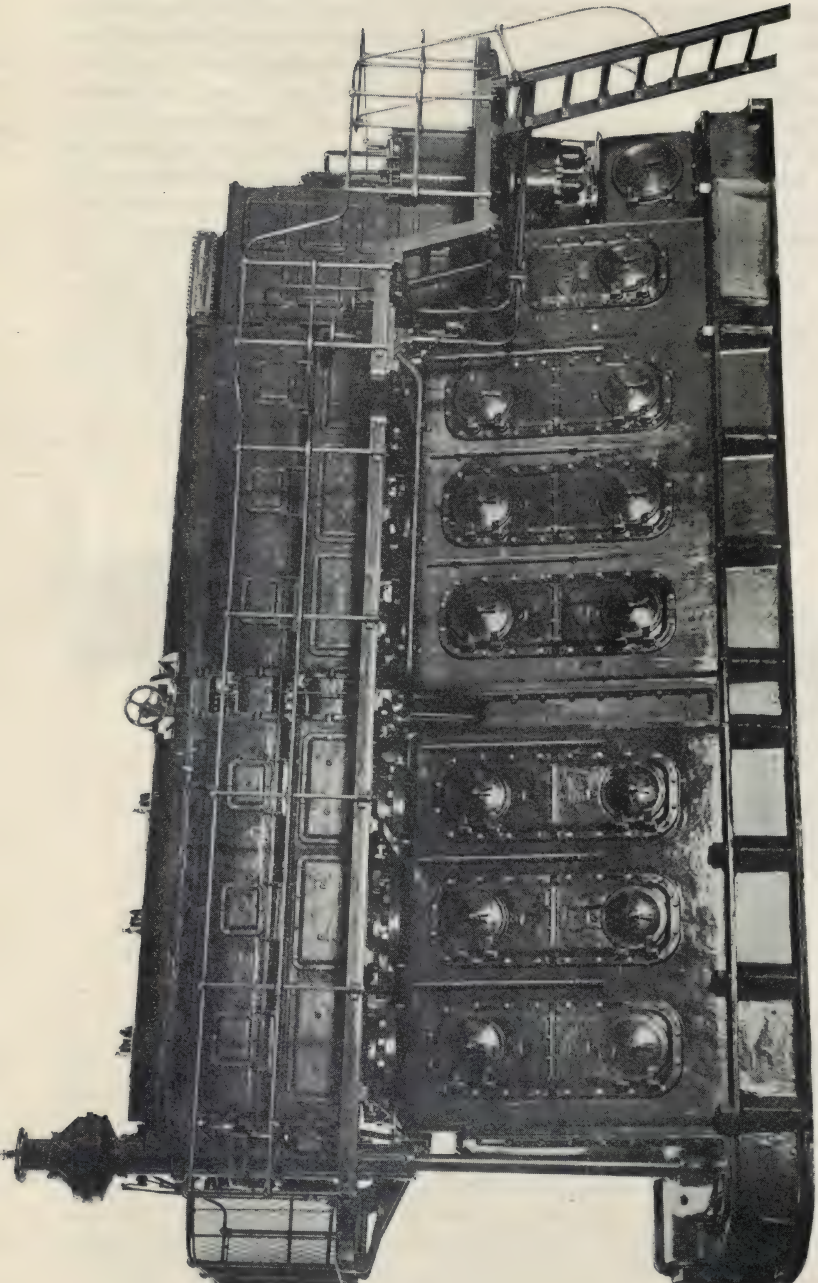
A machine of this type is very rugged and capable of exerting an enormous pull, cases being known where a 25 H. P. windlass like this has pulled a 2-inch steel anchor chain in two without damage to itself. Straight rheostatic control with overload protective devices is nearly always used, and in this case is all installed in the hollow base and cast iron box, making the machine a complete self-contained unit.

Allan-Cunningham's Cargo Winches, as illustrated in cut, are carefully

designed to correspond with the extra heavy work required of a machine of this kind. The service requirements of the cargo winch are very severe, as they not only have to stand up to severe labor when loading or unloading ship, but must withstand all sorts of weather conditions, from excessive heat to deluges of salt water. For moderate sizes, the hollow base type shown is a great favorite, consisting of a series motor with magnetic disc brake spur geared through pinion, intermediate shaft to main gear bolted to the drum, with dynamic lowering controller and resistors mounted in the base. The base is made thoroughly watertight and ventilating doors are provided for use when the winch is operating.



Typical Allan-Cunningham Cargo Winch



1125 B.H.P. Stationary and Marine Two-Cycle Busch-Sulzer Diesel.

CHAPTER X.

DESCRIPTION OF DIESEL ENGINES

BUSCH-SULZER MARINE DIESELS

The Bulsch-Sulzer Engine Company of St. Louis, Mo. was the original, and from 1898 to 1911 the only American Manufacturer of Diesel Engines. Their 23 years of Diesel building and their partnership affiliation with the well known Swiss firm, Sulzer Freres of Winterthur. Switzerland, places them first among the best known American Builders of Diesel Engines.

In following explanation of this well known engine a clear conception will be gained on maintenance and operation of Diesels, in particular the two-cycle constructed.

The engine utilizes, directly in its cylinders, any heavy liquid fuel ranging from kerosene to coal tar.

Pure air, with which the engine cylinder is filled, is compressed by the upward-traveling piston to a pressure of 450 to 500 pounds. Its temperature increases, due to this compression, to approximately 1,000 degrees Fahrenheit. At or near the upper dead-center of the piston the fuel is sprayed into the cylinder, gradually and in a finely nebulized condition. The fuel is gasified and ignited by the heat of the compressed air, without any supplementary means; it burns during the first part of the piston down-stroke, after which the hot gases in the cylinder continue to expand and perform work on the piston, until they are exhausted from the cylinder.

The rate at which the fuel is injected into the cylinder is so adjusted that its ignition and combustion takes place without explosive violence, and with substantially no change of pressure; for which reason this type of engine is sometimes referred to as "Constant pressure" type, to distinguish it from the "Constant volume" type, such as gas, gasoline, and hot bulb engines, and engines improperly designated "Semi-Diesel," in which combustion takes place substantially without increase in volume, and therefore with an explosive-like increase in pressure.

Diesel engines are built to operate on either the four-stroke cycle (briefly, "four-cycle"), or the two-stroke cycle (briefly "two-cycle") system.

In a four-cycle engine, four strokes of the piston, or two revolutions of the crankshaft, are required to complete the cycle of operations. These operations are illustrated in Figures—1 (a) to (d), each of which is accompanied by a description.

Figure 1.

(a) First stroke (admission stroke). Piston travels down; admission valve open; cylinder is being filled with pure air.

(b) Second stroke (compression stroke). Piston travels up; all valves closed; air in cylinder is being compressed.

(c) Third stroke (power stroke). Piston travels down; fuel valve open at top dead-center, but closed at fraction of stroke; gases expand.

(d) Piston traveling up; piston covers exhaust ports; scavenging valve closed; air in cylinder is being compressed.

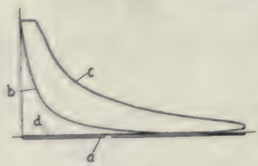


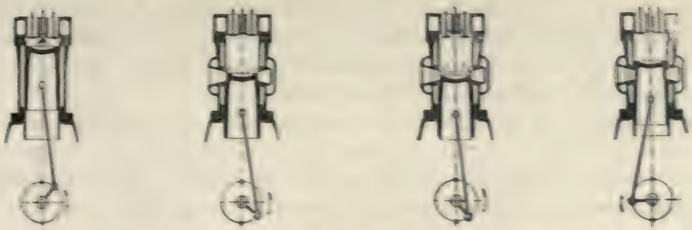
Figure 2.

Figure—2 shows a typical indicator diagram from a four-cycle Diesel Engine, on which diagram the lines representing the various operations are lettered to agree with Figure—2. The constant pressure combustion, which is unique in the Diesel engine, is obvious from this diagram.

In a two-cycle engine, two strokes of the piston, or one revolution of the crankshaft, are required to complete the cycle of operation. These operations are illustrated in Figures 3 (a) to (d), each of which is accompanied by a description.

In the selection of engines for the propulsion of ships, there are practical points which must be considered, namely **Reliability and Simplicity, Economy in Operation, Ease of Operation, Accessibility and Ease of Repair, Space Requirements, and Weight.**

The most important elements which effect the **Reliability** of a Diesel engine are the combustion space in the working cylinder, and the parts which surround this space. Very high temperatures occur in this space, necessitating that the surrounding parts be so constructed that excessive heat stresses will be avoided. This is a compratively simple problem in the case of such parts as the cylinder liner and piston as these are of symmetrical circular form which expand uniformly; but it is much more difficult in the case of the cylinder heads which must contain valve ports or openings. A head with more than one opening is not symmetrical and the increase of the quantity of such openings increases the difficulty of casting the head without shrinkage stress. Increasing the number of the openings also increases the difficulty of cooling, resulting in failure of uneven expansion.



(a) Piston traveling down; all valves and ports closed; fuel injected and burnt during first part of stroke; gases expand.

(b) Piston traveling down; piston uncovers exhaust ports; burnt gases escape through exhaust ports, reducing their pressure to atmospheric.

(c) Piston traveling down; exhaust ports still uncovered; scavenging valves open; air under light pressure enters through scavenging valves, and blows gases out of cylinder.

(d) Fourth stroke (exhaust stroke). Piston travels up; exhaust valve open; burnt gases are expelled from cylinder.

Figure 3.

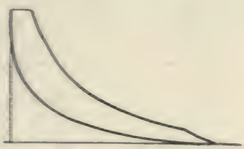


Figure 4.

Figure 4 shows a typical indicator diagram from an ordinary two-cycle Diesel Engine.

Figure 5 shows a sectional plan and elevation of the Busch-Sulzer two-cycle cylinder head and the same views of an equivalent four-cycle cylinder head of modern design. It is obvious from this diagram that the two-cycle cylinder head can safely withstand higher temperatures and the engine may therefore, without sacrifice of reliability, operate continuous with higher mean pressure than any engine having numerous valves in its head.

The principal elements effecting **Economy** in the operation of a Diesel engine are: the cost of the fuel, the quantity of fuel consumed, and the cost of maintaining the engine in operation.

In the case of stationary engines of small and medium capacities, which can be built on the four-cycle system without incurring serious risks of breakdown, it is cheaper to use a good grade of fuel which can readily be obtained in the small quantities required by such engines, than to incur the greater labor expenses of removing the deposits formed by cheaper, low-grade fuels, and correcting the other detrimental effects of such fuels.

In the case of large engines for marine uses, it not only is cheaper but may also be absolutely necessary to operate with low-grade fuels, the first cost of which is lower than that of higher grades where both are obtainable, while they can be obtained in the required quantity in

localities in which it would be difficult or impossible to get the higher grade at any price. For such engines the ability to use a low-grade fuel is, therefore, of primary importance, and in this respect the advantage of the two-cycle engine cannot be questioned.

The absence of exhaust valves, which would be gummed up by asphaltum, and destroyed by sulphur, make it possible to successfully operate the two-cycle Diesel engines with poor grades of oil fuel obtainable in almost any quarter of the globe and at low prices.

Although **Ease of Operation** is important under all circumstances, it applies with special force to marine engines, in which it is essential that starting and reversing be accomplished with ease, rapidity, and absolute certainty.

The starting of a marine engine must be possible and sure with any and all positions of the cranks. To accomplish this at all a four-

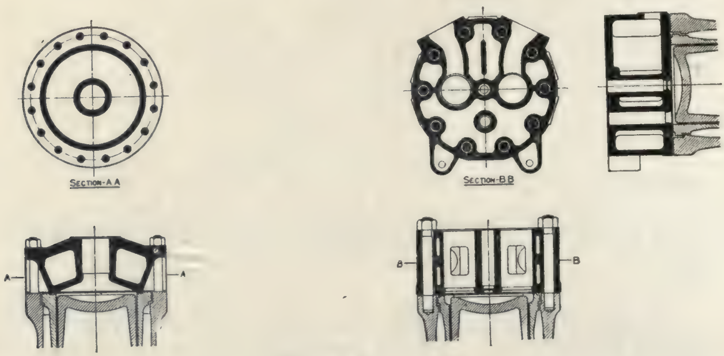


Figure 5.

Two-Cycle (Busch-Sulzer Diesel). Cylinder Head

Four-Cycle (Standard Construction). Cylinder Head

cycle engine must have at least six cylinders; while a four cylinder two-cycle engine has a starting facility equal to that of an eight cylinder four-cycle.

To reverse the two-cycle requires the maneuvering of only the light starting and fuel valve and gears; while the four-cycle requires, in addition to this the maneuvering of the relatively massive and heavily spring-loaded admission and exhaust valves and gears. All moving parts—pistons, connecting rods, crankshafts, flywheel, and line-shafting—which must be brought to rest, restarted, and accelerated, are much lighter in the two-cycle.

The more favorable conditions of crank angle, weight to be handled, and resistance to be overcome, of the two-cycle minimize the dangers of false starts in either direction, and of slowness in the handling of the ship.

In ability to carry overloads, also, the two-cycle has the advantage over the four-cycle. Overloads are emergency requirements, purchased at the price of imperfect fuel combustion, and the consequent deposit of carbon in the combustion spaces. Such deposits cause more trouble with the exhaust valves of a four-cycle, than with any other part of either system of engine.

Unfortunately, every machine is liable to mishaps, and it is important that parts injured by such mishaps may be quickly repaired or replaced. Correct methods of manufacture insure interchangeability, and correct designs **Accesibility**; the other factors which effect replacements are simplicity and ease of demounting and reassembling.

Comparing a four-cycle engine with the two-cycle, it will be found that the cylinder head of the four-cycle has fuel, starting admission, and exhaust valves and their cages mounted on it, and bulky admission and exhaust headers attached to it, in addition to the small fuel, starting air, and water piping; whereas the cylinder head of the Busch-Sulzer two-cycle type has only the fuel and starting valves and their combine cage mounted on it, and the fuel, starting air, spray air, and water piping attached to it.

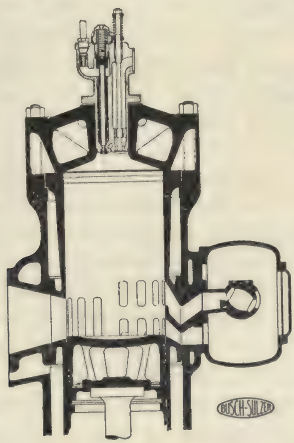


Figure 6. Busch-Sulzer Cylinder. Showing Two-Cycle Scavenging Sulzer System.

Space requirements in the engine room is a more important consideration. For the same power and speed, whatever practical number of cylinders is selected, the two-cycle engine for similar service occupies less space than the four-cycle.

Engines may be built of almost any **weight** per unit of power. Slow-speed stationery Diesel engines weigh as much as 350 to 400 pounds per B. H. P., while high-speed Diesel engines for propelling submarine boats weigh as little as 45 to 50 pounds per B. H. P. In comparing the weights of engines, therefore, the first consideration must be their respective speeds and purposes.

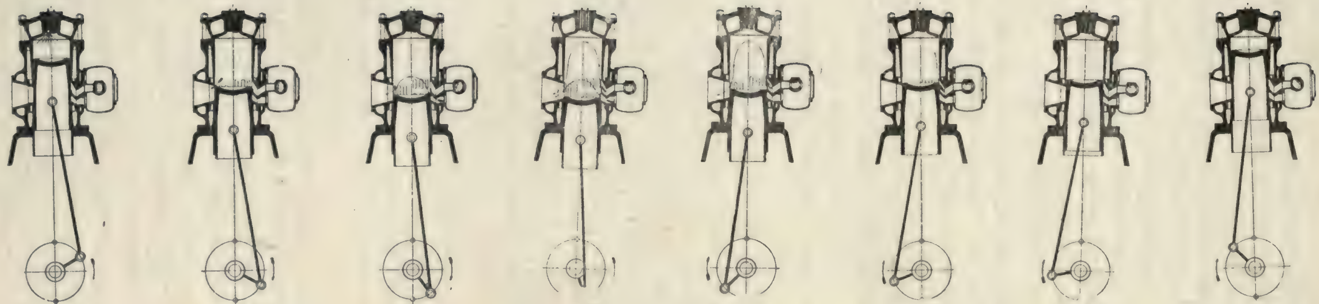
Engines of the same speed and power, and for the same service, vary within wide limits in weights, according to their design, but if of substantially equal ruggedness, the two-cycle Diesel weighs from 20 to 30 per cent less than the four-cycle. Any reduction in weight which is obtained by a reduction of rigidity and safety, should be scrupulously avoided.

Scavenging comprises two functions: the clearing of the previous combustion, or burnt gases, by means of a current of air, and the supply of the air charges necessary for the next combustion. The Sulzer scavenging system used on the Busch-Sulzer Diesel comprises a safe and simple method of thorough scavenging. It utilizes port-scavenging, but employs two tiers, instead of only one tier of ports. The piston uncovers the upper tier of scavenging of ports before, and the lower tier after, it uncovers the exhaust ports, but the communication between the interior of the cylinder and the scavenging-air supply or receiver, through the upper ports, is controlled by a timed and mechanically operated valve, which remains closed until the exhaust ports have been uncovered long enough to reduce the pressure of the gases in the cylinder to nearly atmospheric; after which this valve is opened, while the piston uncovers the lower scavenging ports; a rapid and thorough purging is then effected with complete safety against a blowback into the scavenging receiver.

Upon its return stroke, the piston first covers the lower scavenging ports, and then the exhaust ports; the upper scavenging ports and their valve remain open, enabling the scavenging air to fill the cylinder at full scavenging pressure before the communication is shut off by the piston. Obviously a blow-back of exhaust gases into the cylinder cannot occur; furthermore, the double tier arrangement and proper form of scavenging ports insure a clearing out of such thoroughness that substantially no burnt gases remain in the cylinder—analyses have shown that this residue does not exceed 3 per cent. The weight of air compressed is thus substantially 100 per cent of the weight of a cylinder full at atmospheric pressure, and it is possible to perfectly consume the full quantity of fuel.

The effectively directed streams of scavenging air cool the cylinder more evenly than is possible with ordinary port-scavenging. The complete cycle of the Sulzer scavenging and charging system is shown in figure 7 (a) to (h).

In general, the Busch-Sulzer two-cycle marine Diesel may be described as follows: The engines are vertical, four or six cylinders, single-acting, crosshead type, two-cycle; giving one power stroke per cylinder per revolution of the crankshaft. The injection air compressor is directly driven from a crank on an extension of the main crankshaft. The scavenging air pump is directly driven in the same manner as the compressor except for the larger twin units for which turbo-blowers are employed to supply the scavenging air. The engines are designed for heavy duty service with all parts readily accessible for inspection and adjustment. Workmanship is of the highest class, jigs and fixtures are used in machining processes, so that all parts of the same kind and size are interchangeable.



(a) Piston covers all ports; scavenging valve is closed; gases in cylinder expand. During first fraction of this stroke fuel valve is open and fuel injection and combustion take place.

(b) Piston has uncovered upper scavenging ports, but scavenging valve remains closed; piston is uncovering exhaust ports, burnt gases escape through exhaust ports, reducing their pressure to atmospheric.

(c) Piston is uncovering lower scavenging ports; scavenging valve begins to open; air under light pressure enters through scavenging ports and blows burnt gases out through exhaust ports.

(d) Piston has uncovered all ports; scavenging valve is open; scavenging proceeds.

(e) Piston has covered lower scavenging ports; exhaust ports remain partially uncovered; upper scavenging ports remain uncovered; scavenging valve remains open; scavenging is almost completed.

(f) Piston has covered exhaust ports; upper scavenging ports remain uncovered; scavenging valve remains open; cylinder is being charged with pure air at light pressure.

(g) Piston has covered all ports; scavenging valve begins to close; compression of air in cylinder begins.

(h) Piston approaches top dead - center; scavenging valve is slowly closing; compression of air in cylinder is almost complete.

Figure 7

Iron casting are of grades which experience has shown to be especially adapted to withstand stresses, heat, or wear as demanded by the service for which each is intended. Abrupt changes of section and excessive accumulation of mass are avoided. The steel castings possess carefully determined properties, and are thoroughly annealed. Forgings are of grades of steel which comply with rigid specifications.

Bed Plate and Crank Case: The bed plate is built up of sections, of strong, medium-soft cast iron. It is provided with ample flanges, planed on their undersides, for rigid bolting to the foundation. It comprises an oil-collecting trough with bridges between each two cylinders, which bridges contain bored seats for the main bearing shells.

The crank case is oil and gas tight, and of the enclosed type; it is built up of sections, of the same quality of cast iron as the bedplate. It is rigidly bolted to the top of the bed plate and provided with large covers, readily removable for inspection and adjustments, making all parts inside the crank case easily accessible. The covers carry hinged inspection doors, for convenience in making inspections, while the engine is in operation, without throwing oil.

The crank case carries the crosshead guides, and the cylinder jackets are bolted directly to its top.

Working Cylinders: Each working cylinder consists of two main pieces—an outer jacket which carries all axial stresses; and a liner, which constitutes the running barrel.

The lower end of the outer jacket is bolted directly to the crankcase; the upper end is provided with studs to hold the cylinder head. Openings at the front and back provide for the scavenging air and exhaust connections.

The jacket is of the same quality of cast iron as the bed plate and crank case, and is furnished with hand holes for the inspection and cleaning of the cooling water spaces.

The liner is of special hard, close-grained cast iron, particularly adapted to its service. It is provided with slots or ports in its wall, for the admission of the scavenging air and the discharge of the exhaust gases. The upper end seats on a shoulder in the jacket, making an absolutely water-tight joint.

The center belt, or the portion of the liner at the scavenging and exhaust ports, is turned to fit a bored seat in the jacket, and is provided with packings to make water and gas-tight joints above and below ports.

The lower end of the liner carries the oil-wiper rings, and passes through the bottom flange of the cylinder jacket, where a stuffing box is provided to make a water-tight joint between the outside of the liner and the lower end of the cylinder jacket.

The entire construction allows free expansion of both parts. The space between the liner and the jacket constitutes the water jacket.

The upper face of the liner is provided with a groove, into which the tongue of the cylinder head registers, making an absolute gas-tight joint.

Cylinder Heads: The cylinder heads are of special medium hard cast iron. They are of simple, symmetrical design, the head containing only central opening of relatively small diameter, to receive the combined fuel valve and starting valve cage, thus insuring freedom from casting and heat stresses, and greatest resistance to all working stresses. The heads do not contain any scavenging or exhaust valves. Ample and unobstructed water-jacketing is provided. In this design of head the ring of relatively cool metal surrounding the hot central portion—common to all other designs—has been eliminated.

The cylinder head is rigidly bolted to the top of the cylinder jacket, with a registered fit on the cylinder liner. The underside, or combustion space side, of the head is concave, forming, in conjunction with the concave top face of the piston, a symmetrical combustion space of ideal shape.

The cylinder heads are provided with hand holes for the inspection and cleaning of the cooling water spaces.

Valves and Valve Gear: Scavenging air enters and the exhaust gases are discharged through the ports in the cylinder wall, which ports are opened and closed to the cylinder by the piston uncovering the ports on the down-stroke and covering them on the up-stroke, near the lower end of its stroke.

On the scavenging side of the cylinder there are two tiers of ports. The upper tier is controlled by a timed rotary scavenging valve, driven from the vertical shaft of the engine; the lower tier has a free opening into the scavenging air receiver. This patented arrangement insures perfect scavenging and the complete charging of the cylinder with pure air, while the scavenging valve is out of range of the hot gases.

The fuel valve and starting valve are carried in a common water-cooled cage, located in the central opening in the cylinder head, so that the valves work in a vertical position.

The camshaft, carrying the cams for operating the valves, extend in front and along the tops of the cylinders, in an entirely enclosed casing, and is driven from the crankshaft, at engine speed, through a pair of helical gears at the lower end of the vertical shaft and a pair of bevel gears at the upper end. This drive is located at the flywheel end of the engine, and taken off the main crankshaft on the flywheel side of the first journal, where it is least subjected to torsional irregularities which might effect the operation of the gears and the governing of the engine.

The toothed gears work in oil, and are enclosed in oil-tight housings.

Reversing Gear: The engines are provided with double sets of starting and fuel cams, and the necessary levers and gear to permit the direction of rotation of their crankshafts to be promptly reversed. The gear is air-operated and suitable interlocking devices are provided to safeguard the engine against being started or reversed with the gear in improper position.

Safety Overspeed Governor: The engines are fitted with suitable overspeed governors, which prevent racing, by cutting off the supply of fuel to the cylinders when the speed exceeeds a pre-determined limit, for which the governors may be adjusted. The supply of fuel is automatically re-established as soon as the speed of the engine falls to normal.

Fuel Pump: The fuel pump is of the multiple plunger type (one plunger for each cylinder), operated from the vertical shaft. The function of this pump is to deliver to each cylinder the quantity of fuel necessary to maintain the desired speed and develop the required power. The amount of fuel delivered is determined by the seating point of the fuel pump suction valves, which point is controlled automatically by the governor of the stationary engine, and by hand from the control levers of the marine engine.

The fuel piping is provided with visible overflow valves to free the lines from any accumulated air, which would interfere with prompt starting.

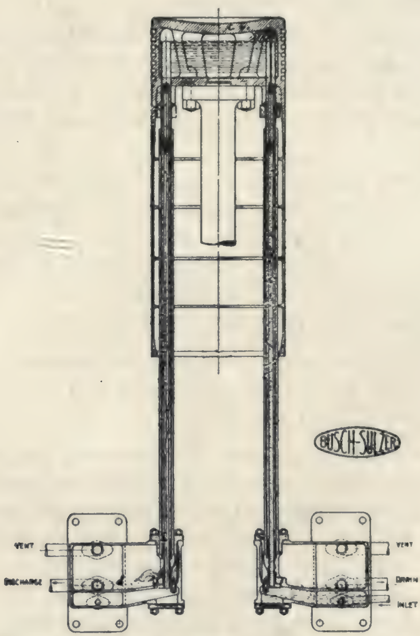


Figure 8. Busch-Sulzer Piston Cooling.

Pistons: The piston proper is short, merely long enough to accommodate the piston rings, as all guiding is performed by the cross-heads. It is provided with a water jacket immediately under its upper face. The piston rod is attached to the lower flange of this piston, by means of studs. The piston is of special, hard, close grained cast iron, best suited to resist the heat and working conditions to which it is subjected. The piston rings are single-piece rings, of a grade of cast iron which will retain its springing qualities until it is worn out.

The piston is water-cooled by a patented arrangement. The water is injected into the cooling chamber in the piston head, and conducted away from same, by a system of telescopic tubes arranged on a principle which avoids swing joints, while preventing oil and water leakage.

Immediately below the piston is fitted a skirt, the sole function of which is to cover the scavenging and exhaust ports in the cylinder wall. The skirt is of hard, close-grained cast iron.

Piston Rods: The piston rod is of forged open-hearth steel. The upper end is provided with an integral flange, for attachment to the piston; the lower end is forked, to connect to the crosshead pin .

Crosshead Pins: The crosshead pin is a high-carbon, open-hearth steel forging. The central part of the pin forms the bearing for the connecting rod. To each side of this bearing is bolted the forked end of the piston rod. The ends of the pins carry the crossheads. Shims are provided between the crosshead pin and the end of the piston rod, to permit the adjustment of the compression in the cylinder.

Crossheads: The crossheads are of cast iron, with babbitted bearing faces. They are of the double, central guide type. No loose or adjustable pieces are attached to them; adjustment being provided by shims under the stationary guides, bolted to the crank case.

Connecting Rods: The connecting rods are of forged open-hearth steel, with marine type crosshead and crank ends. The end bolts are of special soft steel, designed to avoid localized stresses and to resist crystallization.

The crosshead pin boxes and crank pin boxes are of cast steel, babbitt lined, adjustable by means of shims. Dovetailed grooves are machined for anchoring the babbitt in these boxes, and tinned before pouring the babbitt, thus making an absolute bond between babbitt and box.

Crankshaft: The crankshaft is in three sections, each section made from a single open-hearth heat-treated steel forging. Each main section carries two cranks for a four-cylinder engine, or three cranks for a six-cylinder engine, and is provided with integrally forged flanges at both ends. The two main sections are interchangeable with one another. The third section carries the cranks for driving the scavenging pump and air compressor, and is provided with an integral flange to bolt to the main section. All corners are carefully filleted. The shaft is bored to permit examination of the material, and to afford passage for the lubricating oil.

The shaft is made to specifications especially covering the class of steel and manufacture required for this service. Material, dimensions, and construction of the crankshaft are approved by Lloyd's Register of Shipping.

Main Bearings: The main bearings are cast iron cylindrical shells, in halves, lined with babbitt anchored in machined and tinned dovetail grooves. The bottom half-shells are fitted and scraped into the board seats in the bed plate; the top half-shells are fitted into bored seats in the bearing caps. Between the two halves shims are provided for ad-

justment. The shells are securely held in place by the main bearing caps, which are fitted and rigidly bolted to the bed plates.

The seats for the bottom half-shells are absolutely lined up before the shells are put in. After the shells are placed in the seats, they are scraped to the crankshaft, to exact alignment.

Flywheel and Extension Shaft: The flywheel is carried on an extension shaft, connected to the crankshaft by means of a solid-forged coupling flange. The flywheel rim is provided with teeth for barring over the engine. The extension shaft is arranged to couple the thrust.

Scavenging Pump and Receivers: The scavenging pump for providing low-pressure scavenging and charging air for the working cylinders, is mounted vertically on the crankcase, at the opposite end from the flywheel, next to the forward working cylinder and in line with same. It is directly driven from the crank on the extension to the main crankshaft, and is provided with crosshead and guides similar to those of the working cylinders.

The suction and discharge valves are of a patented, simple, automatic "shutter" type mounted in cages. These valves are identical in size and design, and are interchangeable. No springs or plates subject to flexure are used.

The intake side of the pump is provided with a valve chest, arranged so that the scavenging air may, if desired, be brought from outside of the engine room. The discharge side is provided with a valve chest, with connections to the scavenging-air receiver, which, in turn, provides the connection to the working cylinders.

The scavenging-air receiver is of cast iron. It extends along the front of the engine and is bolted to faces on the working cylinder jackets. It provides a firm support for the valve gears, cam-shaft bearings, and casings. A pressure relief is fitted to the receiver.

The suction and discharge valve chests are fitted with large covers, for access to the valves. The scavenging-air receiver is provided with covers for access to the rotary scavenging valves.

Air Compressor: The air compressor, for providing compressed air for fuel injection and starting, is mounted vertically on the crankcase, at the forward end of the engine, in line with working cylinders. It is directly driven from a crank on the extension to the main crankshaft. The compressor is three-stage, water-jacketed, and provided with adequate intercoolers and aftercoolers, to keep the air at a low temperature. The piston is of standard differential trunk type, with patented removable piston pin housing. The compressor valves are without springs and are easily removable for quick inspection and cleaning. The second and third stage valves and seats are of a specially heat-treated alloy steel, found by experiment the best to resist wear and breakage.

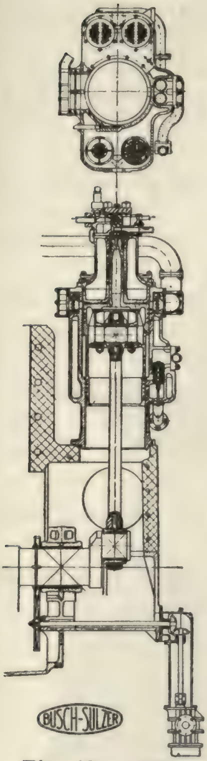


Fig. 10 — Cross Section Through Compressor.

Each stage of the compressor is protected against excessive pressure by a safety valve of special design and ample capacity. A regulating device is provided, to adjust the injection air pressure to suit the operating conditions.

The air coolers are constructed to afford ready access for inspection and cleaning, and are provided with sufficient oil and water separators and drains.

Air Starting System: The engine is started by means of compressed air, furnished by the injection air compressor. The engines are provided with air starting on all cylinders. and one-half of the cylinders are started, followed by the other half.

Air Tank and Piping: The injection and starting air piping is extra heavy, annealed and tested seamless drawn steel tubing, provided with special high-pressure fittings.

Adequate injection air and starting air storage tanks are furnished. Each tank is provided with a special shut-off valve and proper drainage. The starting air tanks are so connected up that they can be charged from the compressor without in any way interfering with the operation of the engine.

The air tanks are of seamless drawn steel; manufactured, tested, inspected, and stamped in accordance with L. C. C. Shipping Container Specifications No. 3-A, and comply with the requirements of Lloyd's Register of Shipping. Each tank is provided with a fusible safety plug, to relieve the pressure in case of fire.

Water Cooled System: The air compressor, cylinder heads, cylinders, pistons, fuel and starting valve cages, exhaust manifolds, and the oil-coolers, are provided with arrangements for efficient water cooling. There are no hidden water overflow and by-pass connections.

All cooling water is discharged into accessible open funnels, and individual outlet pipes are provided for each cylinder, so that the temperature of the discharges from the various engine parts may readily be observed.

Lubrication: The general lubrication is a pressure system providing all main bearings, crank pins and cross-head pin bearings, cross-heads, vertical shaft thrust bearing, and lower helical gears with a continuous supply of cool, clean oil under pressure.

The oil, after passing thru the bearings, is collected in the bed plate, and flows thru a twin filter to a displacement type pump, which

forces it thru a cooler, from which it is again delivered to the bearings, at a pressure of 10 to 20 pounds. A safety valve is provided on the oil pressure pipe; also by-pass connections, for the regulation of the pressure.

All camshaft bearings are provided with oiling rings. The cylinders, including the cylinders of each stage of the compressor, are oiled by a multi-feed pressure type oil pump. Oil cups are provided, where necessary.

Barring Gear: A suitable barring gear is provided for turning the engine over; although the four-cylinder engines of this type do not require barring into a starting position, as they readily start at any crank position.

Platform, Stairs, Railing: The necessary platforms, stairs, and railings to give access to all parts requiring attendance for operation, are provided.

Fuel Oil Service Tank and Filters: A suitable and adequate fuel service tank is furnished, to contain a supply of fuel for the engine.

A twin fuel filter is provided, to remove foreign matter from the fuel before it reaches the engine.

Accessories: Pressure gauges are provided for the air compressor, air storage tanks, lubricating oil, and piston cooling water.

Thermometers are provided in the lubricating oil lines, at points before and after the cooler.

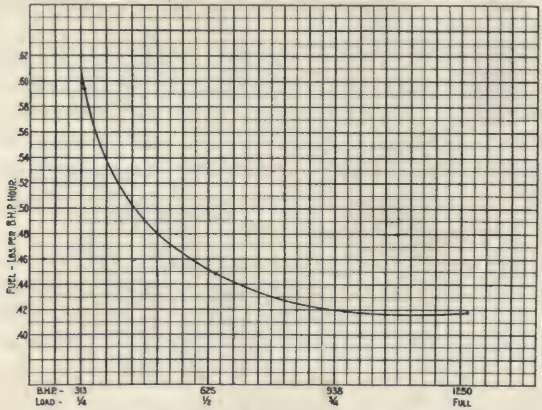


Figure 10.

Curve Showing Fuel Consumption—1,250 B.H.P.—Sulzer Two-Cycle

Under test conditions, and at equal loads, the four-cycle engine may consume 6 to 8% less fuel than the two-cycle. In actual practice this advantage is, however, apparently smaller, due to the fact that the large two-cycle engine may be more safely operated continuously at full load and best efficiency than may the large four-cycle, and also due to the fact that more complete combustion is obtained in the two-cycle at fractional loads, due to the higher temperature in the cylinder.

THE DOW DIESEL ENGINE

The Dow Full Diesel Type Marine Oil Engine is of the vertical multi-cylinder design, built in sizes from 320 B. H. P. to 1000 B. H. P. It follows the four-cycle principle. The engine is direct-reversible and embodies features of improved methods in Diesel operation.

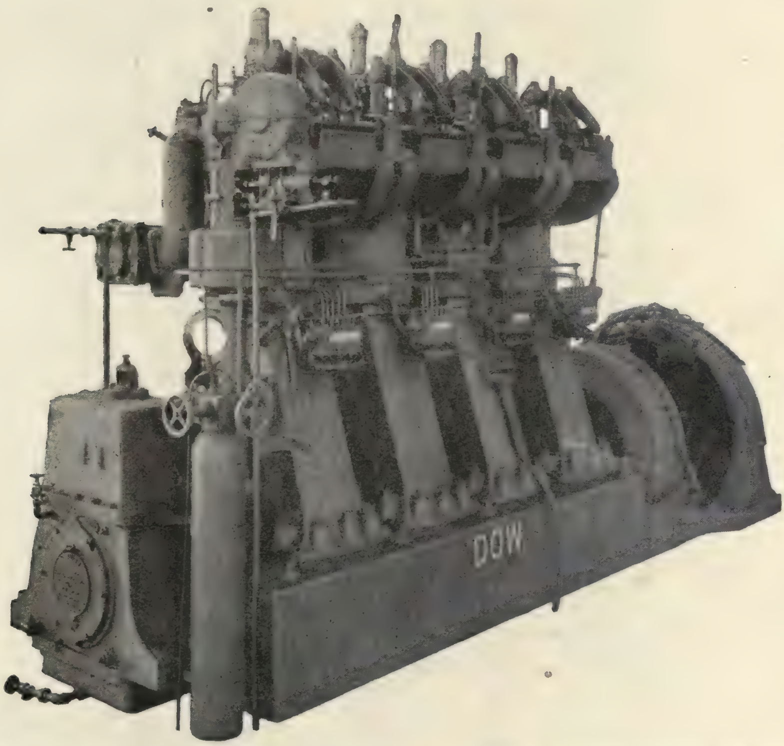


Figure (a) Full View of Dow Direct-Reversible Marine Diesel Engine.

As every respective engine has a special feature characteristic in Diesel manufacture, or a design by which it in many cases tends to be classified in a type of its own, still following the laws as laid down in Diesel prime movers, so has the Dow engine distinctive departures from the usual types of Diesels.

In particular, the designers have laid stress on the importance of equal distribution of fuel. As will be noticed in Figure (a), each set of three cylinders are supplied in common with one fuel pump instead of individual pumps for each cylinder, which permits of an accurate and even distribution of oil for each cylinder.

The output of the fuel pump is controlled by the usual system of governor control, the governor being of the centrifugal type. In similarity to the general principle prevailing on Diesels, the governor, which operates directly on the fuel pump, regulates the supply of fuel oil to the injection valves in proportion to the load on the engine, at all times maintaining the pre-determined speed in revolutions per minute of the propeller.

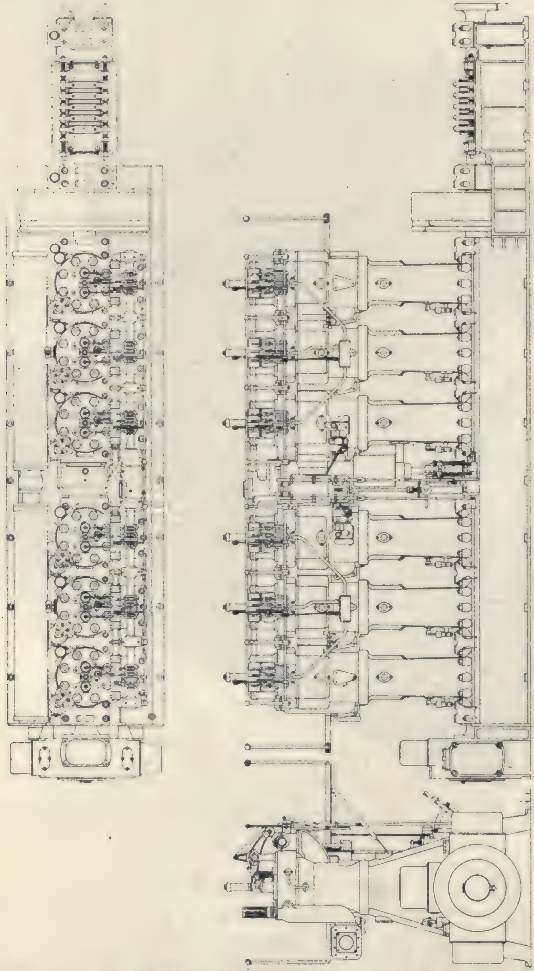


Figure (b) Diagram of Side, Top and Stern usual method in Steam Marine Engine. Note the similarity in the Thrust Arrangement to the usual method of Steam Marine Engine.

The governor is entirely enclosed and mounted on the vertical driving shaft directly geared to the engine crankshaft. It is provided with hand-regulating attachment so that the speed of the engine may be varied by hand while the engine is in operation.

This novel arrangement adds greatly to the accurate performance of the engine, in particular on long voyages where heavy seas are experi-

enced and racing of engine are the results, very often causing complications with consequential injury to the engine.

The cylinder heads are of box section and water-jacketed. Air, fuel, exhaust and starting valves are all contained in the cylinder head, their operating levers being securely supported by bearings mounted on and bolted to the heads.

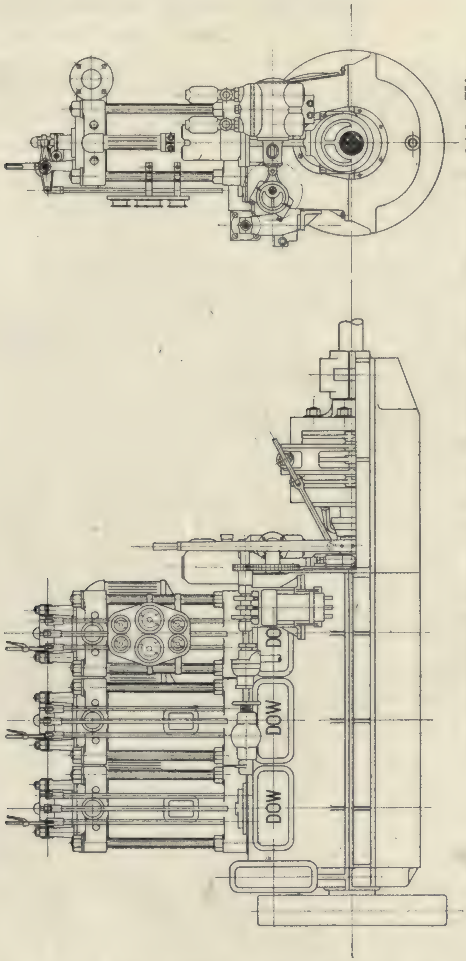


Fig. (c) Diagramatical View of Dow Diesel Engine. Front and Side View

Special care has been given to the construction of all valves and valve seats, which are of the removable type, allowing them to be readily withdrawn from the cylinder head for inspection and re-grinding when required. All the valves are opened by the action of steel levers rolling on hardened cams. The cams are accurately set and keyed in place, insuring positive action. The valve gear is of the cam type throughout, the cam-shaft being driven by machine-cut spiral gears. All levers for oper-

ating cams are provided with case-hardened rollers and pins. Special care is taken in the design of the exhaust valve lever, so that the exhaust valve and seat may be removed without disturbing the valve gear.

Reversing Mechanism: Briefly described, the reversing of the engine consists in the changing over from one set of cams to another; the work being performed by an air-driven ram. The operator simply moves a small hand lever to accomplish the reversal position. This operation is immediately followed by the movement of the main control lever to starting air position, where it is allowed to remain during one or two impulses. At the completion of this operation this same lever is moved to running position. The whole above-described maneuver is accomplished in five seconds from full speed ahead to full speed astern position.

The air-driven cams referred to above automatically lifts the cam rollers clear of all cams, then slides the cam shaft to the desired position and returns the cam rollers to the cams. The control lever is securely interlocked with shifting mechanism, preventing any false move on the part of the operator.

The thrust block is of the standard marine horseshoe type, is equipped for water cooling. It is securely bolted and doweled to the main engine bed-plate, insuring perfect alignment.

In illustration (c) an ideal engine is shown for tugboat service or yachts, or such smaller types of vessels employed on coast-wise trade. As will be seen, this engine is equipped with reverse-gear. This type of engine stands up to heavy duty such engines are very often subjected to.

The Dow Diesel Engine follows the "A" frame construction. The bed-plate, which extends the full width and length of the engine, is of box girder section throughout, reinforced with transverse and longitudinal ribs. The bed-plate carries the main bearing journals and seating for the "A" frame.

Forced lubrication is provided for all cylinders and piston pins, while all main journals and outboard bearings are furnished with ring oilers. All crank pins are lubricated by gravity and centrifugal oilers. (**Note.**—For Compressor, see Section on Compressors.)

THE FULTON DIESEL ENGINE

The Fulton Iron Works Company, in developing the Fulton Diesel Engine, as patterned and designed by Franco Tosi, one of Europe's foremost engineers of Diesel Engines, has produced one of the most efficient and dependable prime movers obtainable.

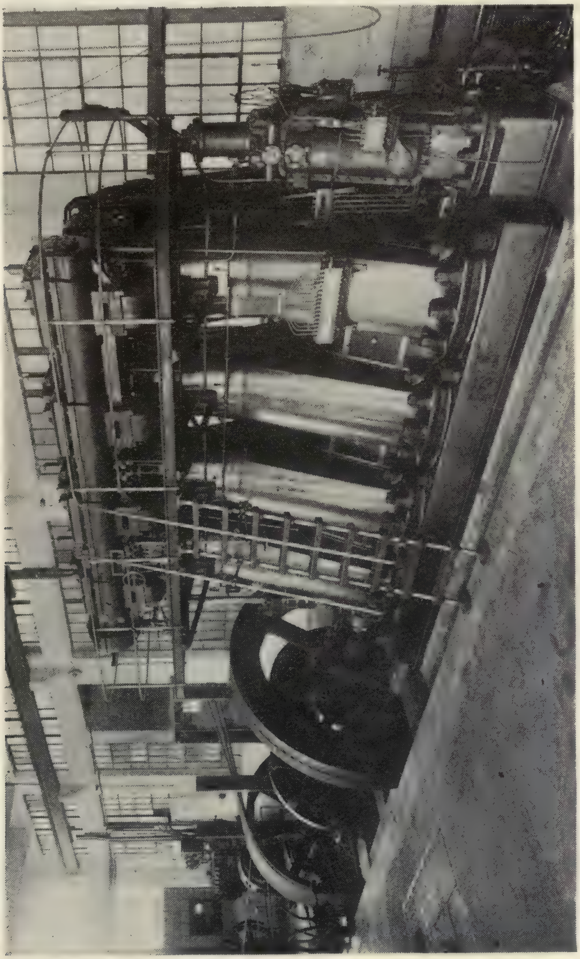


Illustration of 585 B.H.P. Fulton Diesel Engine Driving Flour Mill.

The Fulton Diesel Oil Engines not only possess all the advantages of the typical European oil engine, but has been developed by many refinements and improvements for American service by the engineers of the Fulton Iron Works Company. With following explanation it will be

seen that the improvements embodied in this engine brings it on an equal standing with modern types, in particular as an engine adapted for stationary service.

As will be seen in other pages of this book, the Tosi Diesel engine has been brought to the highest stage of perfection and is used today on exceedingly large ships in merchant service.

Like the Junkers type and many German patents, it has been a product created by years of experiments and brought up to the highest stage of perfection. It is an oil engine operating on fuels of low gravity and in consequence its maintenance is very inexpensive.

The most distinctive features of this engine are found in the strength of their heavy "A" frame construction, the accessibility of all bearings, and the fact that the cylinder liners can be readily removed without dismantling the engine.

The cam shaft is on a level with the cylinder head, and the arrangement of all parts is so simple that adjustments can be made easily and economically, even while the engine is in operation.

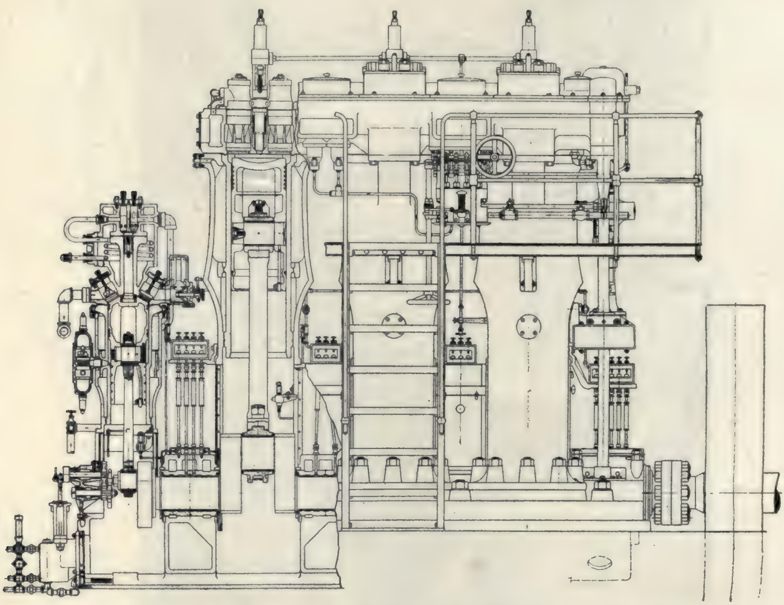


Diagram Showing Full Illustration of Engine with Compressor, Pumps, Lubrication System, Etc.

As the Fulton Oil Engine is of the four-cycle, vertical, "A" frame type, it is not subjected to the strains that are usually found in internal combustion engines of the horizonal types. This design practically eliminates vibration which is an additional factor for long life in the engine.

This engine is equipped with a three-stage compressor. It is directly connected to the crankshaft end and provides the large storage tank with highly compressed air for both starting and for the purpose of fuel injection.

The Fulton fuel pump is of the variable plunger stroke type, with positive control from the governor. The needle valves are under control from operator's stand. The exceptional strong construction of the force-feed pump is of similar design as will be seen in the usual types of Diesel Engine. The necessity in providing proper lubrication method is taken care of in this engine by sight-feed gravity oilers, which supply all bearings with copious quantities of oil.

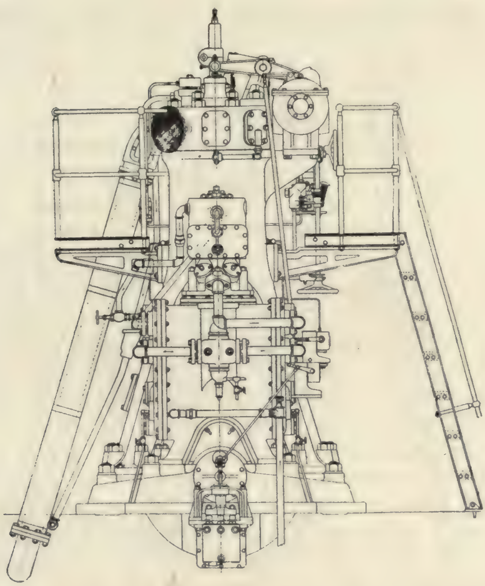


Diagram of Fulton Diesel Engine, Looking from Compressor In. End View of Engine

By means of the Fulton patent starting mechanism the air is delivered to the starting cylinders and at the same time positively locks all fuel oil from the cylinders.

The Fulton design is such as allows of quick, easy and inexpensive replacement of all wearing parts by means of bushing, brasses, etc.

A very advantageous feature in adding to the economy of the engine is the system of oil filtering. All the oil that is introduced into the engine is handled by a three-plunger pump at the compressor end of the crankshaft. The center plunger pumps fuel oil from the storage tank to a gravity tank. The outside plungers pump clean lubricating oil to a gravity tank and the dirty oil from the pump to the filter. All the lubricating oil that flows into the pump is run into a settling tank, then through an oil filter where the oil is filtered through bone black. This permits the lubricating oil to be reclaimed in a large measure and none but clean oil to return to the engine. By the use of bone black for filtering, the oil is completely revivified. By the above means of lubrication great economy has been obtained. By actual operating figures obtained from a 500 B.H.P. 3-cylinder engine, driving a 350-kw. alternator, it is found that:

1 gallon cylinder oil serves	8,283.3	kw. hr.
1 gallon bearing oil serves	7,455	kw. hr.
1 gallon compressor oil serves	74,550	kw. hr.

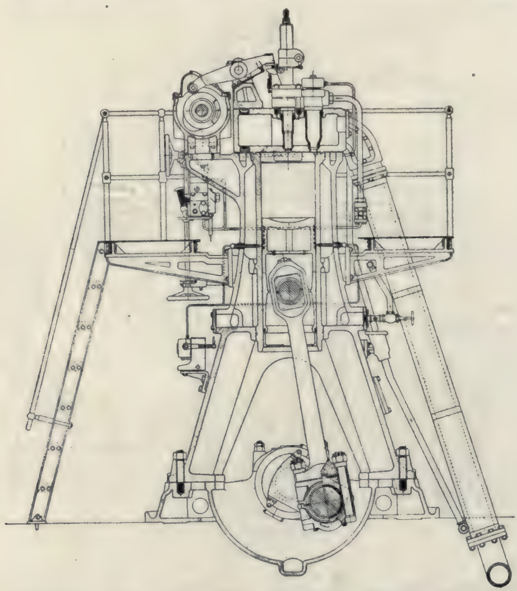


Diagram of Fulton Diesel Engine. Good View

The governing of the Fulton Diesel oil engine is accomplished by a Jahn's type governor, driven from the vertical shaft. This governor acts directly on the fuel pump of the engine so that any change in the load is almost instantly compensated by a quick response from the engine. There is no lagging or "hunting."

The old method of driving the governor by the vertical shaft was to place the governor directly on that shaft. This necessitated using a

large governor, running the same speed as the shaft. It was found that every vibration imparted to the cam shaft by the action of the cam was transmitted directly to the vertical shaft and hence to the governor, causing undue wear. To eliminate this undue wear on the Fulton engine, the governor was offset from the vertical shaft, and driven by flexible gears. It was also found that by this method a smaller governor could be used and run at higher speed, thus giving a better regulation and smooth operation with any of the undesirable vibrations.

The cylinders are lubricated in a simple manner. The cylinder oil flows by gravity to a Richardson Phoenix sight feed oiler and from there individual brass piping conducts the oil to four points in each cylinder wall. These points are located a little below the center of the piston when it is at the top of its stroke. Thus oil is not carried to the firing chamber by the piston rings nor carried to the pump by the wiper ring.

The piston pin lubrication is one of the special positive feed systems on the Fulton engine. The oil line leading to the sight feed oilers for the main bearings is tapped just below the shut-off valve. From that point a smaller pipe with a single sight feed adjusting valve runs down to a point inside the "A" frame just below the bottom edge of the cylinder liner. At this point, fastened to the "A" frame by bracket, is the lower half of the piston pin oiler. This oiler is a special apparatus, having a small cylinder containing a plunger that has a hole bored through its center. The plunger is held by a spring from beneath. The operation of this mechanism is such that any oil introduced into the lower end of the cylinder is forced through the hole in the center of the plunger by any downward action of that plunger. This downward action is accomplished by the upper half of the piston pin oiler (consisting of a check valve), which is rigidly attached to the inside of the engine's piston skirt at its bottom edge, striking the oiler's cylinder plunger at the end of each downward stroke of the piston. By an adjusting screw on the oiler, the stroke of the oiler's plunger can be regulated from about one-fourth of an inch to any small fraction of an inch, depending upon the amount of oil desired to reach the piston pin. The upper half of this oiler (the part attached to the engine's piston) is screwed to the end of a pipe that runs up the inside of the piston and direct to the piston pin. Thus the oil flows by gravity to the lower half of the piston pin oiler and from there is forced direct to the piston pin by a positive feed, controlled by regulating the stroke of the oiler's plunger and by the amount of oil the operator allows to flow to the oiler.

The reliability, continuity of operation, maintenance, etc., of the Fulton Diesel Oil Engines, are on a parity with those of a high grade steam plants of corresponding capacity. In fact, the results achieved show that the maintenance is even very much less than that of a steam plant of corresponding capacity, with the added advantage of low labor and fuel costs. The investment charges are approximately the same on this market as for a fully and modernly equipped steam plant, which, of course, takes into consideration the entire property, including building.

THE NORDBERG-CARELS DIESEL ENGINE

The Nordberg-Carels Diesel Engine is an improved development of the well known Carels engine of Belgium, the Nordberg Manufacturing Company of Milwaukee, Wisconsin, having secured rights to build this engine from Carels Brothers, of Ghent, Belgium. The engines are of the two-cycle type.

The engines range in size from 750 to 3000 B.H.P., and in units of from three to six cylinders. The larger engines have cylinders of 500 B.H.P. each, being the largest size built in America. The engines are of the vertical, heavy duty type, with crossheads and open frame. All engines operate at relatively slow speeds, a feature which promotes long life of the engine and a minimum of time lost in shut-downs for repairs or overhauling. This is of special importance in marine installations, reducing expensive delays in port and serious breakdowns at sea.

All the above engines have cam actuated scavenging valves in the cylinder heads. A smaller engine has been developed by the Nordberg Manufacturing Company, of the port scavenging type; that is, air is introduced into the cylinder by the movement of the piston, uncovering a series of ports in communication with the scavenging pump.

Inasmuch as the Carels engine follows the two-cycle construction, the method of scavenging follows similar systems adopted on modern engines of this respective construction. The older system of providing scavenging valves in the cylinder head appears to be abolished in most two-cycle Diesels, ports being provided at the bottom of the cylinder, uncovered by the piston in arrangement with auxiliary valve-controlled air ports usually just above the main ports.

Following is a tabulation of the several sizes and types built by the Nordberg Manufacturing Company:

Type	B. H. P. Sea Level Rating	Type Scavenging	Speed Stationary Units
3 V.E.	330	Port	225
4 V.E.	440	Port	225
5 V.E.	550	Port	225
6 V.E.	660	Port	225
3 E.G.	750	Valves	180
4 E.G.	1000	Valves	180
5 E.G.	1250	Valves	180
3 F.H.	1500	Valves	120
4 F.H.	2000	Valves	120
5 F.H.	2500	Valves	120
6 F.H.	3000	Valves	120

In the case of marine engines direct connected to the propeller shaft, the above engine speeds are reduced to suit lower propeller speeds, the stroke being lengthened to compensate. In electric marine drives the above standard speeds are maintained.

Two-cycle engines are particularly well adapted for continuous operation with low grade fuels. This is, because there are no exhaust valve seats subjected to intense heat or to become pitted or corroded with heavy oil residue, necessitating frequent shutdowns for cleaning and refitting of valves.

The following is the fuel consumption based on oil of 18500 B.T.U. per lb., using any quantity of fuel free from water:

Load	Full	¾	½
Lbs. Oil per B.H.P. per hr.	0.45	0.47	0.51

One B.H.P. is secured from about 8,000 B.T.U., which corresponds to over 750 B.H.P. hours per barrel. Lubricating oil consumption ranges from .001 lb. per B.H.P. hr. for large units to .0015 lbs. for smaller units.

Following is a tabulation of cooling water required.

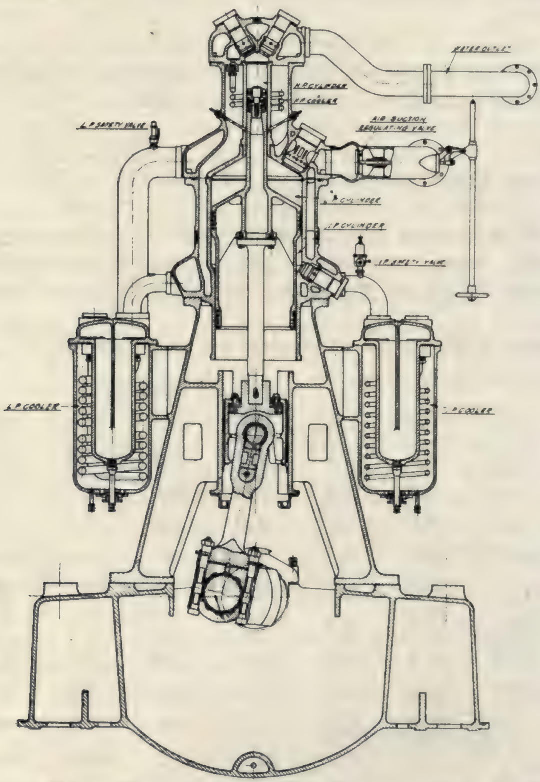
Inlet	Discharge Temperature							
Temp.	95°	100°	105°	110°	115°	120°	125°	130°
50	7.2	6.5	5.9	5.4	5.0	4.6	4.3	4.1
55	8.1	7.2	6.5	5.9	5.4	5.	4.6	4.3
60	9.2	8.1	7.2	6.5	5.9	5.4	5.	4.6
65	10.8	9.2	8.1	7.2	6.5	5.9	5.4	5.
70	13.0	10.8	9.2	8.1	7.2	6.5	5.9	5.4
75	16.2	13.0	10.8	9.2	8.1	7.2	6.5	5.9
80	21.6	16.2	13.0	10.8	9.2	8.1	7.2	6.5
85	32.4	21.6	16.2	13.0	10.8	9.2	8.1	7.2
90	---	32.4	21.6	16.2	13.0	10.9	9.2	8.1
--	---	---	32.4	21.6	16.2	13.0	10.8	9.2

Nordberg Diesel engines include direct connected generator units for central municipal and industrial power stations, also direct connection to Nordberg two-stage air compressors, centrifugal pumps, belt drive to ammonia compressors, gear drive to plunger pumps, etc.

Details of Construction: The illustrations show the general design of the Nordberg engine. It will be noted that open frame construction has been adopted on this type of Diesel. This renders the running gear easily accessible for inspection. However, the openings between frames are closed by means of light weight, oil tight, removable guards.

Bedplates are of the closed pit type for collection of lubricating oil. Cylinder barrels consist of removable liners, at the middle of which are

located the exhaust ports. Cylinders, heads, pistons, exhaust pipes, cross-head pins and cross-head guides are water cooled. The absence of exhaust valves in two cycle engines simplifies the construction of the cylinder head and renders the cooling of it more uniform, thus eliminating danger of cracking. This is often the case when the construction of Diesels is such that sudden heat temperatures or lack of cooling seriously effects the plant, causing damage.



Cross-Sectional View Through Nordberg Engine. Note Coolers, Air Suction Regulating Valve, Water Circulating System, Crosshead Guides, Scavenging Valves, etc.

Locating the scavenging valves in the cylinder head considerably strengthens the head and placing scavenging valves as found on the Nordberg-Carels engine makes very effective and uniform cooling.

The use of crosshead allows a larger clearance between cylinder and piston, eliminating all danger of the piston seizing in the cylinder. In particular should this be beneficial when the engine is working under heavy load or prepared to be shut down. The importance of this detail can hardly be over-estimated, as serious accidents have occurred in the so-called "trunk-piston" design.

A special feature of the Nordberg engine is the crankshaft, which is of the "built up" type. The crank pin and crank webs are made from one solid forging and the shaft sections pressed in and keyed.

This construction is highly commendable and adds to the permanency of this engine.

Fuel atomizing, scavenging, and air starting valves are shown in cross-sectional illustration. The fuel valve is of the closed type, the needle of which is closed by means of an outside spring and opened by a cam. This needle is easily removable for inspection without taking the balance of the valve apart. The fuel valve is adapted to handle fuels varying from a very light oil down to high asphaltum such as Mexican fuel oil, and those of similarity on the Pacific Coast of 12° Baume.

Each cylinder is provided with its own fuel pump, which is of the plunger type, eccentric driven from the cam shaft. Any pump can be cut out independently of the others and inspected while the engine is in operation. Means for priming are provided.

Close and accurate regulation is obtained by the use of a sensitive, rigidly constructed type of governor, driven by the cam shaft by means of an elastic drive, making the rotation of the governor uniform and thus increasing its accuracy. The regulation of the engine speed is accomplished by varying the quantity of the fuel oil introduced to the fuel valve. The governor acts upon the fuel pump by passing the fuel in greater or less quantities, according to the power demand.

Air for starting and fuel atomizing is supplied by a three stage single acting compressor, direct connected to the engine and provided with inter-coolers. Compressor valves are of circular plate type, no valve gear being required.

The engines are provided with an automatic oiling system complete with filter, pumps, etc. Cylinders are lubricated by independent mechanically operated oil pumps. In addition to having feeds from the oiling system, the main bearings are of the ring oiling type. A cooling coil is provided in the overhead tank to insure proper cooling of lubricating oil.

A fuel oil filter and fuel heating arrangement is included. Fuel oil passes from the filter to the fuel pumps through a heated header.

McINTOSH & SEYMOUR DIESELS

In the illustration of the McIntosh & Seymour 390 indicated horsepower engine, a type of light but strong construction is shown. In particular the accessibility to vital parts during operation will be noticed. Both frame and base are well ribbed to insure the proper stiffness and strength. On this size, the cylinders and frame are cast in one piece, the cylinders being arranged in sets of three. (Chapter VIII, page 166).

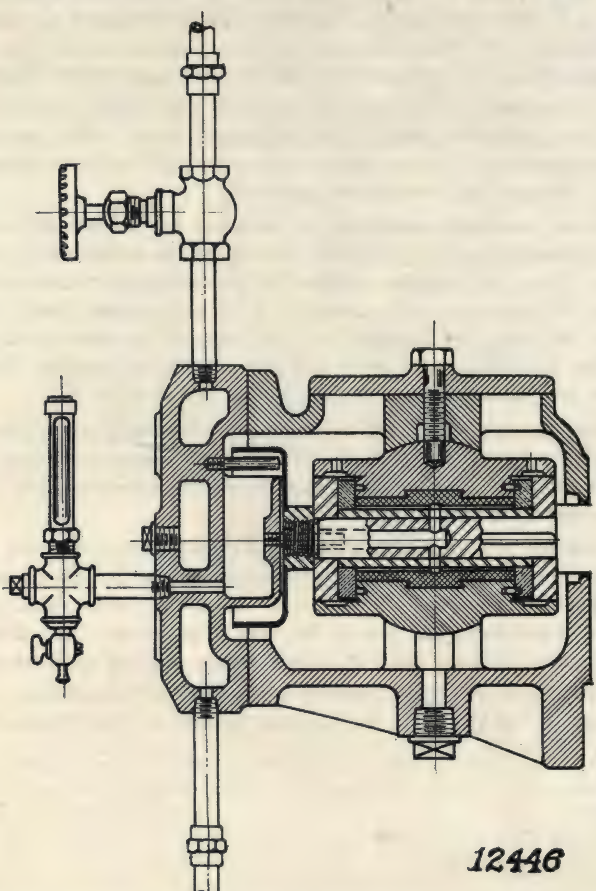


Diagram of Kingsbury Thrustbearing Extensively Used on Medium Sized McIntosh & Seymour Diesel Engines With Great Results

The control consists of only two levers, one to control the fuel and starting air, and the other the maneuvering. The first half of the movement of the fuel lever on the quadrant controls the stroke of the fuel pumps, which regulates the amount of fuel delivered to the cylinders and

consequently the speed of the engine; the second half of the movement is used in starting. This opens the relay valve and admits the starting air to the cylinders. The reversing is accomplished by turning the reversing wheel. This can be done in a short time, as from three to five seconds.

The attached compressor is driven from the forward end of the crankshaft. This compressor is three stage and the air is thoroughly cooled between the stages and after the high stage, by inter-and after-coolers.

A Kingsbury thrust bearing is attached directly to the base of the engine. This thrust bearing is of the late type and is noted for its reliability and efficiency.

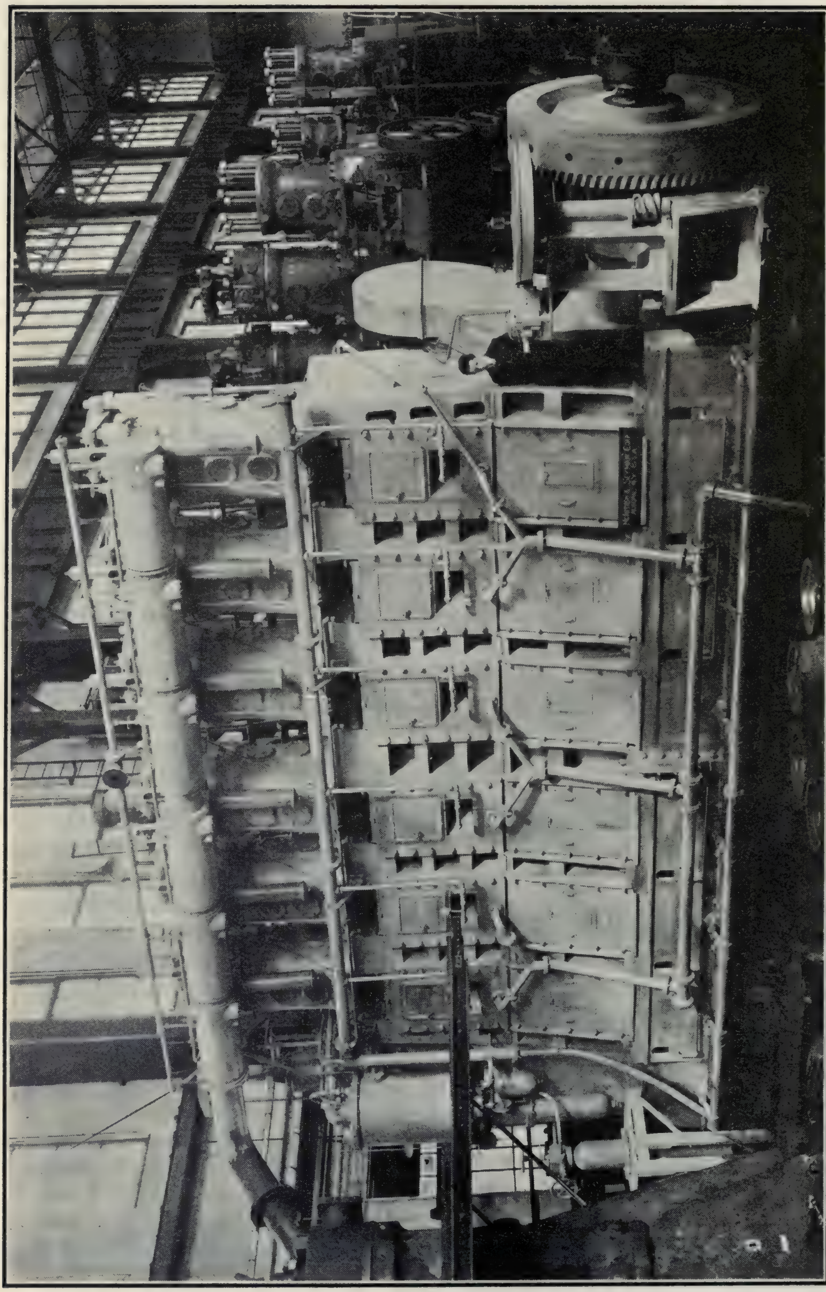
In the illustration of the 1200 I.H.P., such as installed on the Motor Ship "Kennecott", it represents the cross-head type of Marine Engine.

The bases have the same arrangement of bearing girder that the smaller engines have, which gives the base great stiffness, so imperative in marine engineering. The engine is arranged with individual frames, each frame being of the same general arrangement as the corresponding section of the box frame on the smaller engines.

These individual frames are held tight together on the front by the guide plate, which is attached at the top and bottom with fitted bolts and on the back by a tie plate also bolted by fitted bolts; this tie plate being arranged to carry the cooling water pipes for the pistons. All of these crosshead type engines are arranged for water-cooled pistons, as when cross-heads are used, the piston cooling piping can be so arranged that there will be no chance for any cooling water to get into the lubricating oil. A diaphragm is fitted below the cylinder with a packing ring around the piston rod so that no water can get into the working parts, and this also prevents any excess lubricating oil from getting on the cylinders, so that pressure lubrication can be used on the working parts if it is so desired.

With the cross-head type engine, the maneuvering is accomplished by means of air cylinders or an electric motor in designated construction. The fuel being shut off, the maneuvering gear first removes the cam rollers from the cams, then moves the cam shaft endways in similarity to the smaller types, placing the other set of cams under the roller; then it puts the rollers back on the cams; the gear being so arranged that when this is accomplished the fuel lever can be operated and the engine started.

It will be interesting here to give some minor details on the accomplishment of the McIntosh & Seymour engines on the "Kennecott." The ship was built by the Todd Drydock & Construction Company, in 1921, for the Alaska Steamship Company. It has a length of 345 feet, a beam of 49 feet 6 inches, and a loaded draft of 22 feet. This vessel is equipped with two McIntosh & Seymour Heavy Duty cross-head type Diesel marine engines of 1,200 indicated horsepower each, with a nominal full load speed of 140 R.P.M.



Rear View of 1200 I.H.P. McIntosh & Seymour Diesel Engine. Installed in M/S Kennecott."

All the auxiliaries on this vessel are electrically driven, in similarity to the M. S. Solitaire, illustrated in other pages, there being no boiler on the vessel. Electricity is furnished by two McIntosh & Seymour 100 B.H.P. Diesel electric generating sets, one of these being in operation at sea, which drives all the auxiliaries as well as taking care of such heating as there may be. Two of these engines are used in port for operating the winches. This vessel has a deadweight tonnage of 6,560, and it averages a little over 10½ knots at sea, fully loaded, and has a fuel consumption of about 10½ tons per day of regular 16 gravity boiler fuel oil.

It has been found with the electric winches on the "Kennecott," that she can handle cargo 50 per cent faster than with steam equipment and with a fuel consumption when in port of four barrels per day.

Commercially, the "Kennecott" is of interest from the fact that it can make money at rates where a steam vessel has to be operated at a loss.

The illustration of the McIntosh & Seymour Diesel engine gives an excellent view of the modern type of this respective make. While in smaller sizes the trunk piston is preferred, nevertheless in larger classification the cross-head type seems to find greater favor. The general construction of McIntosh & Seymour's engines are exceedingly rigid. The base is made very stiff by arranging the bolting between the frame and the base so as to give a very short, effective length of the bearing girder, giving great strength and stiffness with a minimum depth of base.

The frame is of the box type, with large openings for each crank, and the ribs which form the sides of these openings extend clear across the frame with an arch over the bearing cap, which gives a frame of great stiffness and rigidity with a minimum weight.

On the small sizes, the cylinder jackets are cast in one piece with this box frame, and on the larger sizes the cylinders are cast separately and bolted to the frame.

The air compressor is of the three stage type and is driven from an overhung crank on the forward end of the crankshaft. The trunk piston types of engine have forced feed lubrication for the cylinders, piston pins and the compressor, and have gravity pressure lubrication for the main bearings and crank pins so as to avoid any excess lubricating oil getting in the cylinders.

The smaller sizes of trunk piston type of engines are maneuvered by hand. The fuel being shut off, the first turn of maneuvering hand wheel removes the camrollers from the cams, the next two turns of the wheel moves the cam shaft endways, placing another set of cams under the rollers, and the fourth turn puts the cam rollers back on the cams; the gear being so arranged that until this operation is complete, the fuel lever cannot be moved.

To demonstrate the ease of maneuvering the McIntosh & Seymour engines of the trunk type, we will refer back to this subject. The control consists of only two levers, one to control the fuel and starting air,

and the other the maneuvering. The first half of the movement of the full lever on the quadrant **controls the stroke** of the fuel pumps, which regulates the amount of fuel delivered to the cylinder and consequently the speed of the engine; the second half of the movement is **used in starting.** This opens the relay valve and admits the starting air to the cylinders. The reversing is accomplished by **turning the reversing wheel.** Three turns of the wheel is all that is necessary to completely reverse the engine. This can be done in as short a time as from three to five seconds. A Kingsbury thrust bearing is attached directly to the base of the engine. This thrust bearing is of the latest type and is noted for its reliability and efficiency.

COMPARISON TABLE OF THERMAL EFFICIENCY, OVER ALL

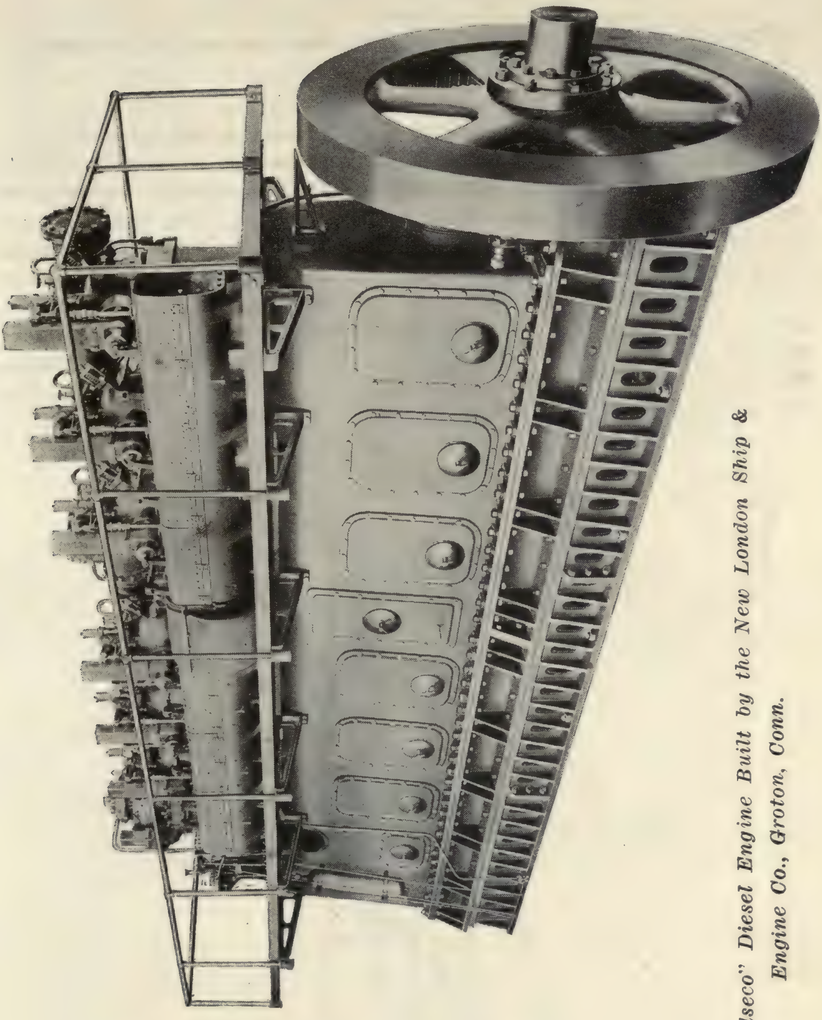
Of Diesel Station and Three Types of Steam Stations, all Operated Under the Same Management.

(McIntosh & Seymour Diesel)

	Plant A	Plant B	Plant C	Plant D
Rating of plant	1050 kw.	1300 kw.	1050	1055
Type of plant	Diesel	Steam Tur.	Steam Eng.	Gas Engine
Number of units	3	3	3	4
Fuel used	Oil	Oil and Coal	Gas and Coal	Gas
Average station factor, per cent:		B. T. U. per KW. Hour		
96.6	11,734	-----	-----	-----
71.8	11,822	-----	-----	-----
49.0	-----	27,000	-----	-----
46.3	13,556	-----	-----	-----
34.0	-----	-----	-----	26.100
32.0	-----	-----	43.200	-----
23.6	18,203	-----	-----	-----

REPORT OF TEST ON McINTOSH & SEYMOUR DIESEL ENGINE (500 B.H.P.) IN A MODERN POWER PLANT OPERATING MANUFACTURING ESTABLISHMENT.

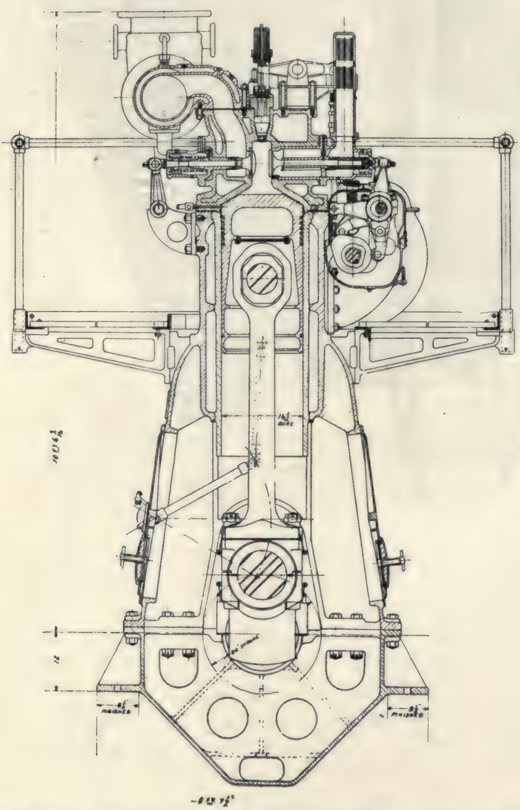
Percentage of rated load	25.6	51.0	75.8	99.6	114.8
Revolutions per minute	168	167	166	163	168
Brake horse power	128.2	255	379	498	574
Time of test (in hours)	½	½	½	1	1
Fuel consump. per B.H.P.-Hr. (lbs.)	.584	.432	.396	.393	.388
Injection pressure (lbs.)	750	775	800	800	900
Exhaust gas appearance	Clear	Clear	Clear	Clear	Clear
Inlet temp. of cooling water (°F.)	69	69	69	69	69
Outlet temp. of cooling water (°F.)	150	150	158	158	158
Temperature in testing room (°F.)	73	73	72	78	79



600-750 B.H.P. "Nelseco" Diesel Engine Built by the New London Ship & Engine Co., Groton, Conn.

DETAILED DESCRIPTION OF NELSECO MARINE AND STATIONARY DIESEL ENGINES.

As will be observed in accompanying illustrations of the 600 B.H.P. Nelseco engine, the six working cylinders of the engine are in line with the single three stage air compressor at the forward end. Forward of the air compressor is the fuel pump and governor. The bedplate and housing are of exceptionally rigid design and construction, the housing being carried right up to the tops of the cylinders and the cylinder-liners forced into the housing with a space between which forms the water-jacket. Detachable cylinder-heads are bolted directly to the top of the housing. All of the valves in the cylinder head are arranged horizontally, and are operated from the camshaft, which is carried in brackets on the side of the housing by means of vertical rocker arms.



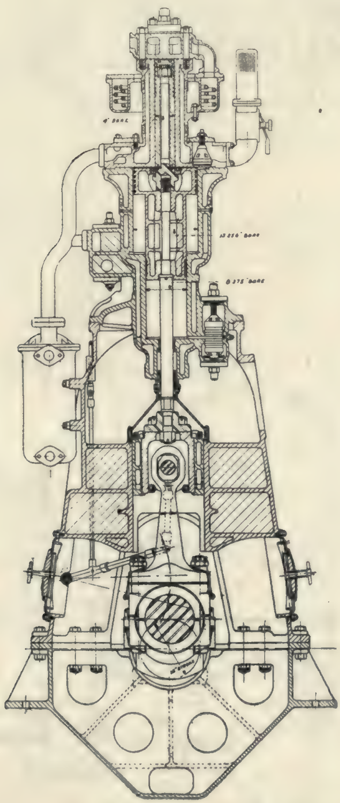
Section Through Working Cylinder of Nelseco 600 B.H.P. Engine.

With this design there is a space of about six inches between the injection valve nozzle and the piston top, which aids good combustion and prevents burning and cracking of the pistons.

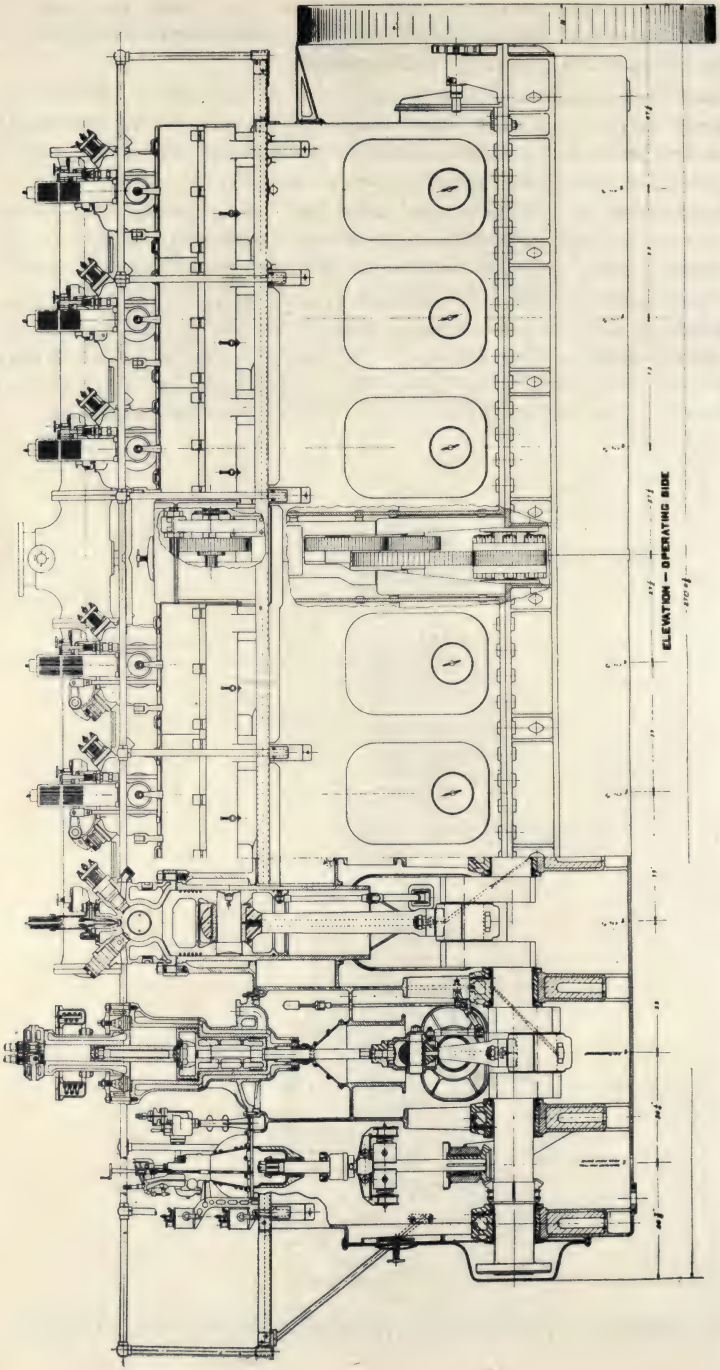
It has been suggested that undue wear of the lower side of the valve-stems may be caused by the weight of the valve being constantly borne at one point, but in actual practice this has not transpired, nor is any detrimental effect anticipated by the designers.

The camshaft and much of the valve gear, are enclosed in a casing and driven by a train of spur-gears from the crankshaft, located in the middle of the engine between working cylinder numbers three and four.

We will mention that the bearings at both ends of the connecting rod, as well as every other important bearing on the engine, are adjustable, which is an important feature in the case of a marine-engine, where shims occasionally have to be taken out. Large openings are provided in the sides of the housing to permit a free access to the crankcase.



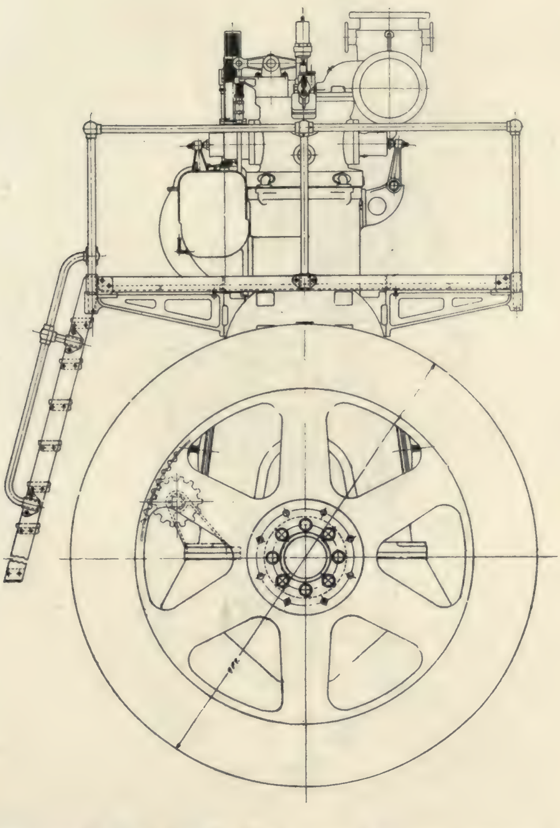
Section Through A. C. Cylinder of 600 B.H.P. Nelseco Diesel Engine.



Diagramatical View of 600-750 B.H.P., 200 R.P.M. "Nelseco" Diesel Engine. An excellent view is allowed to valves, gears, connecting rods, etc.

As already stated, the air compressor is of the three stage type. In this case the piston is driven from the crankshaft by means of a connecting rod and cross head. Advantage is taken of this opportunity to separate the compressor cylinders entirely from the crankcase, providing an accessible machine as well as avoidance of all trouble due to lubricating oil in the cylinders, which was a fault found too frequently in early Marine Diesel engines, and occasionally met with even today.

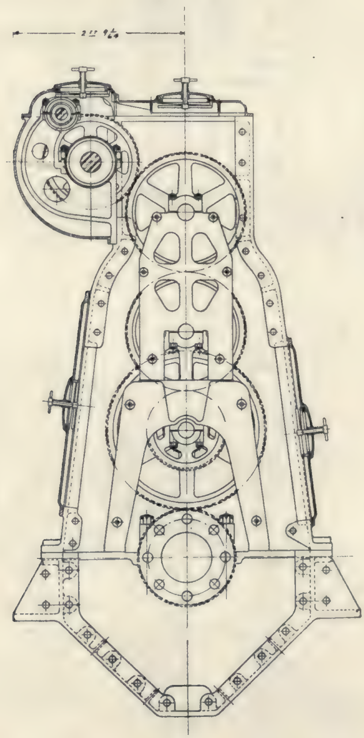
For fuel injection, fuel pumps provide a separate plunger for each working cylinder, and is driven directly from the crankshaft by means of a vertical shaft and suitable gearing. On this vertical shaft is mounted the governor, which is of the constant-speed type for engines designed to drive generators, and of the limit-speed type for direct-connected marine engines. All controls and also the reversing gear for the directly



Nelseco 600 B.H.P., 200 R.P.M. Diesel Engine, looking astern. Note the valve actuating arrangement.

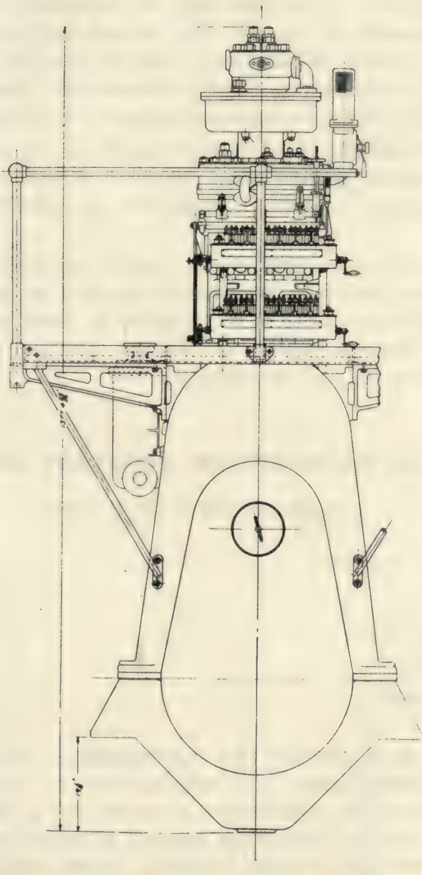
reversible marine engines are located at the forward end of the engine and on the upper platform, that is, at the fuel pump and air compressor end, and here, of course, the operator's station is located. In the case of generator engines, only three cylinders are fitted with air starting valves, but for direct reversible engines it is necessary to fit all of the working cylinders with air starting valves. Starting air for this purpose is taken from storage tanks, and a maximum of 350 pounds pressure is used.

Independently driven circulating water and lubricating oil pumps are provided, as the Nelseco designers consider this the most satisfactory method, but provision is made for fitting direct connected pumps at the forward end of the engine in such special cases as it may seem advisable or necessary. Provision is made for ample water jackets on all parts of the engine which require cooling, and the arrangements of connection areas are such that there is a free flow of circulating water from the inlet to the overboard discharge from the exhaust headed jacket.



Section Through Cam Shaft Gear Compartment of Nelseco 600 B. H. P. 200 R.P.M. Diesel Engine.

For the lubricating of the engine, the system used can best be described by calling it a gravity forced-feed. With this system the lubricating oil flows from the gravity tank, which is at a sufficient height above the engine crankshaft to give the proper pressure to the main bearings, then through holes in the crankshaft to the crankpins, and up the inside of the connecting rods to the wrist pins. The crankcase is enclosed and all surplus oil drains to a suitable formed trough in the bottom of the bedplate from whence it is pumped back through suitable strainers to the gravity tank. All of the important bearings are thus under forced lubrication and a free flow of oil is circulating through them at all times.



Oil Pump Arrangement of Nelseco 600 *B. H. P.* 200 *R. P. M. Diesel Engine.*

The camshaft parts are oiled by splash from oil carried in the bottom of the trough of the camshaft casing. All cylinders and exhaust valves stems are taken care of by mechanical oilers, and the minor valve gear bearings are fitted with oil cups for hand oiling.

For large freighters, passenger vessels and tankers up to about 12,000 tons D.W.C., and 4,000 H.P., this engine is made non-reversible and used in conjunction with electric transmission. In these cases several engines make up the total power required, although any one unit would be sufficient to bring the vessel home at reduced speed in case of accident to the others. At the same time it must be pointed out that a totally disabled modern direct driven Diesel ocean motorship is practically unheard of today.

For Diesel electric drive of a cargo ship requiring 2,000 shaft H.P. (about 2,600 I.H.P.) four of these 600-700 B.H.P. units, each connected to a 400 K.W. electric generator, may be installed in the engine room and these are capable to provide the current for a single electric propelling motor. Control of the latter can be placed on the navigation bridge, or at any other part of the ship independent from the engine room, thus eliminating delays or misunderstandings when maneuvering.

For detailed description on this arrangement the readers are advised to study the chapter dealing with "Electric Propulsion," pertaining to the trawler "Mariner," which is propelled by twin-240 B.H.P. Nelseco Diesel engines.

This engine has been built under Lloyd's survey, and is designed to meet Lloyd's and American Bureau of Shipping requirements in every way. In the following table it is intended to show actual performance of this type of Nelseco engine in runs at constant speed and various powers, illustrating its fuel consumption:

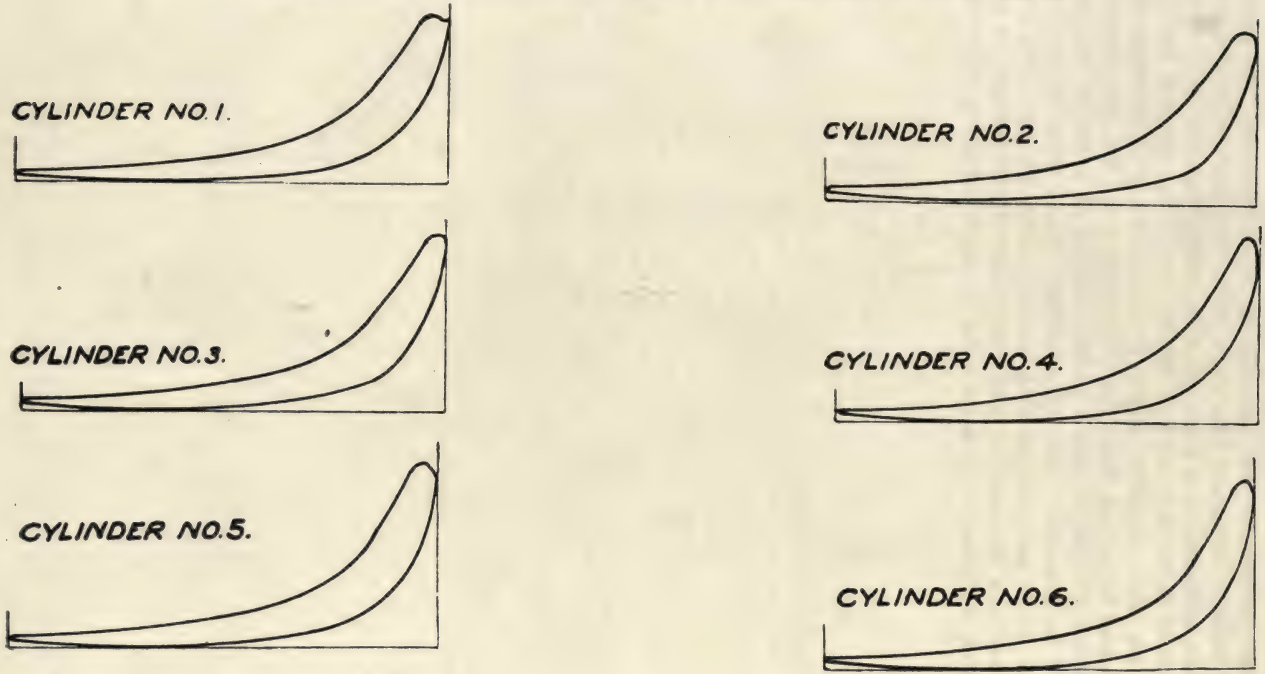
FUEL CONSUMPTION—CONSTANT SPEED

(In pounds per B. H. P. hour)

25 per cent over-load	.42
Full load	.42
Three-fourths load	.45
One-half load	.51
One-fourth load	.67

This table shows the flat fuel consumption curve, which is characteristic of the Diesel engine. To give an idea of the overload capacity of this engine, it can be stated that it carried for a short time a load of 875 B.H.P. with only a very slight amount of smoke showing in the exhaust. This shows the conservativeness of the 600 B.H.P. rating. In fact, the engine is fully capable of developing 10 per cent overload almost continuously. Another run was made with the revolutions vary-

Set of Indicator Cards taken on test of Nelseco 600 B.H.P. Diesel Engine.

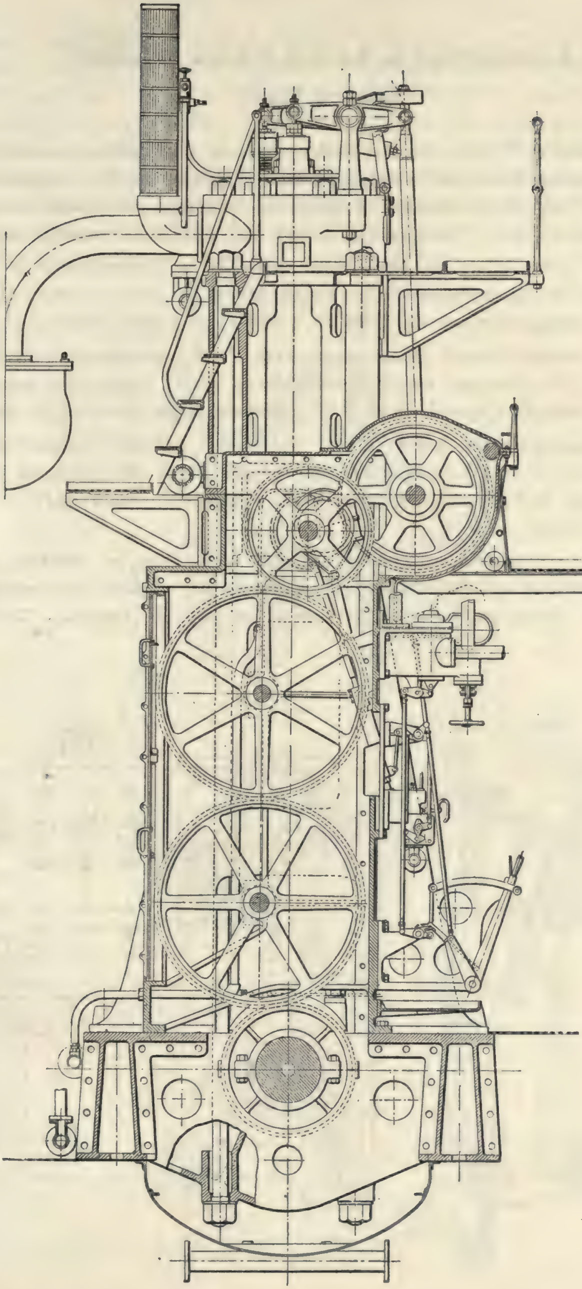


Spray Air = 950 Lbs. per Sq. In.
R.P.M. = 225.

Average Compression Pressure = 480 Lbs. per Sq. In.
Average M.I.P. = 883 Lbs. per Sq. In.

Average I.H.P. = 128.6.

ing as they did when direct connected to a propeller aboard ship, and it was found that with the speed reduced to about two-thirds when the load would be approximately only one-quarter that of full power, the consumption of fuel was only about .52 lbs. per horsepower. The marine fuel consumption curves are thus much flatter than the fuel consumption curve when operating on a generator. The mechanical efficiency of the engine proved to be very excellent indeed, and at full power it was just under 80 per cent. Governor trials were also held and it was found that from full load to no load there was only about a 3½ per cent variation in revolutions. When the load was thrown on or off suddenly there was practically no momentary jump in revolutions except when full load was thrown on or off, and then the first surge amounted to only a few revolutions more than the speed at which the engine finally settled. In fact, the governor performance appeared to be given the service in every way.



End View of Burmeister & Wain Diesel. Note Gear, Valve Arrangement, Engine Control Lever, Etc.

BURMEISTER & WAIN'S DIESEL ENGINES

(Marine Type)

Burmeister & Wain's Diesel engines are a convincing factor of modern development in Diesel construction. Engines of this type may be found in all respective classes of ships of Europe's Merchant Marine as well as of America's. The engines follow the four-cycle construction.

To gain some idea on advanced methods of Burmeister & Wain's engines, a brief description is herewith given of late adopted styles of engines for cargo-carriers of 5,000 to 6,000 tons displacement.

In this type of vessel the average daily fuel consumption is about 25 tons of oil. The average speed is calculated on 11 knots per hour. The engine and propeller speed, 80 R.P.M. Engine room crew of 15 men.

The dimensions are I.H.P. 1,750; shaft H.P. cylinder diameter 24.803 in.; stroke 51.181 in.; revolutions 100; length from aft coupling to front of compressor 10,500 mm.; height from center of crankshaft to top of valves 6,600 mm.

The most interesting features of this new type of engine are the cylinder head and cylinder liner. The heads are nearly square and are supported by distance pieces which stand on the "A" frames. The heads

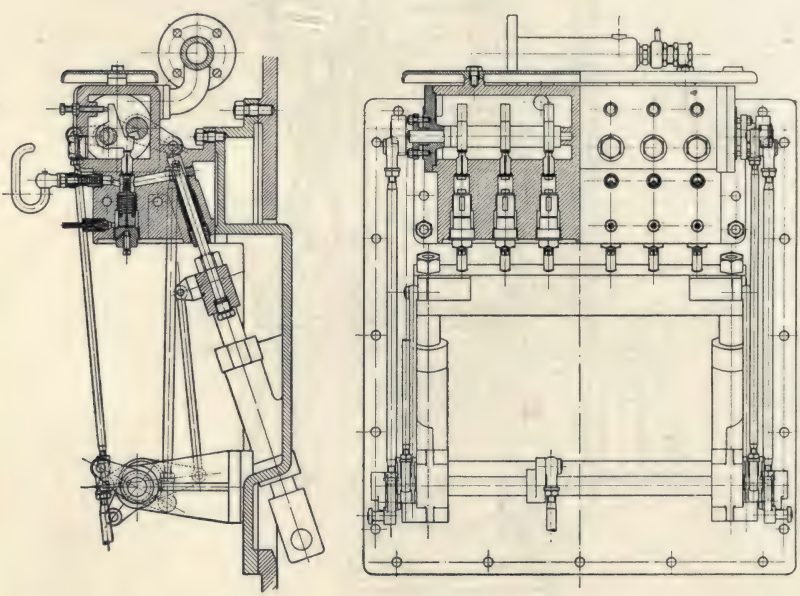


Diagram of Burmeister & Wain Marine Diesel Engine, Showing Front View of Force-feed Fuel Pressure Pump.

are bolted together, three and three. Long bolts pass from the main frame, thru the "A" frames and distance pieces, and between the heads, each nut pressing down on two heads. The long bolts also have nuts at the top of the "A" frames. The liners are bolted directly to the head, without any packing, iron to iron. Outside the liner, bolted to the head with a gasket is the water jacket, which packs off against the liner near the lower end with a rubber ring. This allows the liner perfect freedom to expand. The inner shell of the cylinder head is well braced by four stays. The cooling water enters at the lower end of the jacket and most

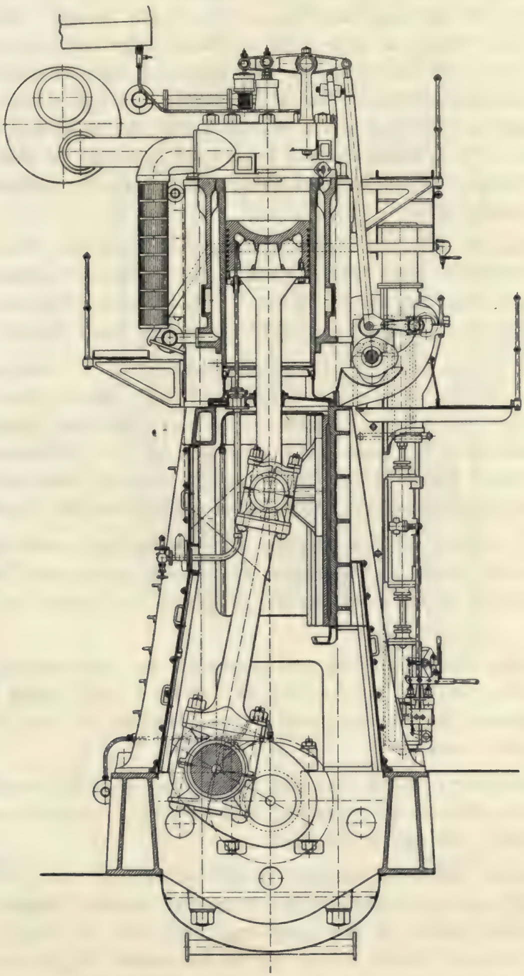


Diagram of Burmeister & Wain Diesel Engine. Cross-Sectional View looking astern.

of it is made to pass into the cylinder head at the opposite side by a baffle-ring inside the jacket. The water space in the head is very large, insuring effective cooling around all valves.

In order that the removal of the piston downward instead of by removing the head, may not be too complicated, it has been made oil-cooled instead of water-cooled, as in previous builds.

To remove the piston, it is brought almost to top center, the cross-head shoe is blocked up, the cross-head bearings taken apart, and the connecting rod lowered by turning the crank almost to bottom center while lowering the upper end of the connecting rod with a chain hoist, so that it rests on the outboard side of the main frame. The outlet and inlet valves are removed, eye-bolts screwed into the piston head, and the piston is lowered through the top plate so that the cross-head pin rests on beams placed across the main frame. It can then be inspected or even removed. The top plate differs from the ordinary construction in that it has a hole large enough to permit passage of the piston, this hole being closed by two semi-circular plates, which contain a scrape-ring for the piston rod.

The frame for the gear-train stands in the center but is separate from the "A" frames. The three fore and the three aft cylinder heads are bolted together with horizontal bolts. This construction allows the engine to weave slightly, too great rigidity having been found detrimental.

The main frame and "A" frame are largely of "I" section. The air compressor is of the latest Burmeister & Wain three stage design, directly connected to the crankshaft at its forward end, having its own base frame bolted to the main frame, though its "A" frame and cylinders stand free. Other parts of the motor are in general Burmeister & Wain standard, with such small refinements as experience has made desirable.

One new feature of this engine which deserves mention takes the form of a metric scale and hand-control wheel regulating the clearance or varying the lift of the fuel valves, enabling the engine to run at very slow speed.

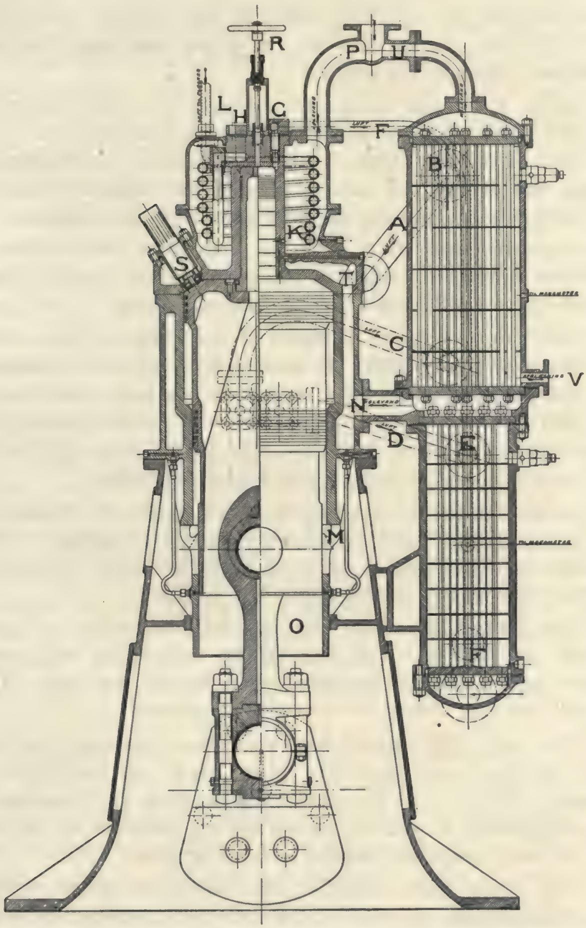
At the after end of the engine room are the lubricating oil pumps, supplying lubricating oil to the engine, while a fuel pump is stationed near the bulk-head for the purpose of pumping the oil from the tanks to the daily service tanks.

It is interesting to note that the Burmeister & Wain engineers strictly adhere to the four-cycle principle, and following reasons are advanced, which impartially we are giving here.

In the Diesel engine, combustion and process of work take place in the same unit, and thereby differs from the steam engine, where the combustion takes place in the furnace. The heat is transferred to the water in the boiler, whilst the work is developed in the engine. It is therefore a well known fact, that if the boiler is forced, furnace and tubes will be affected by the fire and the economy of the plant be annihilated, seeing that a considerable loss of heat is caused by the

products of combustion leaving the funnel without having been cooled down.

The same is the case with the Diesel engines. If too great a quantity of fuel per hour is combustioned in these, the surfaces are too intensely heated, and will become scorched and destroyed, as the heat cannot penetrate the walls to the surrounding cooling water. Also a considerable economical loss is caused, through the temperature of the exhaust gases being too high.



Diagram, Showing Cross-Sectional View of Burmeister & Wain Marine Diesel Engine.

However, in comparing steam-plants with Diesel plants a thorough difference will be found, because in a boiler the useful heat passes to the water through the heating surface, whereas in a Diesel engine, the lost

heat passes through the surface of the cylinder to the cooling water. A standard for the strain of the **boiler** is how many kilograms of coal are burned per hour in proportion to the principal dimensions; likewise the standard strain of the Diesel engine is how many kilograms of oil are burned per hour in proportion to its principal dimensions. **It is immaterial, whether this takes place** in a four-stroke cycle or in a two-stroke cycle system. On the other hand the economy with which the combustion takes place is of the most eminent significance, because the heat, which is not transformed into useful work, i. e., the heat lost, will partly be passed through the cylinder walls to the cooling water, partly disappears in the heat of the exhaust. **It is the lost heat that strains the most important parts of the engine.**

Burmeister & Wains engineers contend that the four-stroke cycle engine is the most economical as its consumption is 15 to 20 per cent less than that of the two-stroke cycle; consequently **the four-stroke cycle is the type that causes the least strain on the material**; the four cycle system is therefore absolutely in their opinion to be adaptable.

Upon entering more fully into the details of the two motor types, the four cycle shows, compared to the two-cycle, in every respect advantages which in concentrated form are the following:

(1) The time the inner surfaces of the cylinder and cover are exposed to the high temperature of combustion, is in the four-cycle motor only half the corresponding time of that in the two-cycle, the cooling of the inner surfaces during the suction stroke is more effective in the four-cycle motor, consequently the four-cycle works as a whole far cooler than the two-cycle at the same development of power.

(2) In the four-cycle the piston speed allowable is higher, and as the mean temperature is lower, the work can be performed with a higher mean pressure. Therefore a four-cycle of the same weight and outer dimensions develops the same horsepower, or more.

(3) The four-cycle engine having no scavenging air pump with appurtenant air receivers nor any larger scavenging air channels, the construction of the engine becomes more simple and easy. Dimensions are reduced, and the engine works more noiselessly.

(4) The four-cycle engine can work more regularly at low revolutions, and is able to go perfectly "dead slow" as well as any marine steam engine. This is not the case with the two-cycle, because the pressure of compression is smaller owing to the pressure of the scavenging air being reduced, when the engine is running slow.

(5) The whole valve gear of the four-cycle works with half the number of revolutions of that of the two-cycle, this gives a softer and more silent running and less wear and tear on the different parts.

(6) For the reason above mentioned not only the additional quantity of oil used in a two-cycle engine to develop one H.P. is lost, but the heat thus produced strains the engine excessively, shortens its life and increases the overhauling costs, so that the additional consumption of fuel oil is in more than one sense lost. This is the case particularly in

large engines of high horsepower, by reason of the large cylinder dimensions requiring proportionately heavier castings and consequently, it is very important not to conduct greater quantities of heat than necessary through the cylinder walls in order not to strain the material.

The details of the marine Diesel engine are designed with the particular requirements of each single part in veiw, and as these are different from those of a marine steam engine, the structure and appearance of a marine Diesel engine are quite diverging from that of an ordinary marine engine.

The Burmeister & Wain Diesel engines follow closely the design of their steam marine engines. This particularly applies to crankshafts, main bearings, connecting rods, crossheads and guide shoes.

The air compressor, by which the air for the fuel injection is compressed up to a pressure of 60 atmospheres, forms a most important part of a Diesel engine. For this purpose the type used is of their own design, combined with the engine.

The engine is direct reversible and may be built in capacities ranging from 200 to 8,000 H.P.

In large types of ships the installation is of the twin-screw system. The two engines work independently. A Diesel ship of large capacity is usually equipped with double bottom tanks, to have sufficient fuel oil to carry it along for 31,000 miles, while the steamer's bunker of ordinary size can only carry enough bunker coal or oil to carry it along for an average of 4,800 miles.

Engines formerly used, built by the Burmeister & Wain Company, were of the two-cycle system, after experimenting with the four-cycle type, the company succeeded in developing a four-cycle engine, which, on the same revolution, accomplished the same effective horsepower, had the same length, the same breadth, but smaller height and weighed 1½ tons less than the two-cycle of a corresponding horsepower. The total weight of the engine in question, was 9 tons; that of the two-cycle 10½ tons. Moreover, the consumption of fuel oil in the four-cycle engine was 17 per cent less. Seven of these engines have been supplied to the Danish Navy.

These engines develop 450 E.H.P. at 500 revolutions per minute, and owing to their lighter weight and elegant construction they are especially suitable for fast running motor boats and yachts.

The engine can be made to other dimensions if required.

Of similar special engines we mention a 600 E.H.P. direct reversible marine engine of 280 revolutions per minute built for the Societe Anonyme John Cockerill, Seraing, Belgium, for river service on the Upper Kongo. This engine is a type somewhat heavier than that aforementioned, but considering the small draft of these vessels and the consequential small diameter of the propellers they are naturally rather fast running.

A new designed type of engine with an extra long stroke and having a corresponding low number of revolutions especially suitable for single-screw cargo ships, are now building. These engines are intended for slow going cargo ships and can, owing to the low number of revolutions, be fitted with large propellers, thereby giving the vessel a particularly good speed in bad weather, as well as a great maneuvering capacity; likewise a good control of the vessel is obtained, which is important when sailing in narrow waters or maneuvering in or out of port.

Owing to their low number of revolutions these engines are of course, a little more expensive to build per H.P. They present on the other hand, such advantages that will quickly pay to use this somewhat more costly engine for the purpose mentioned.

The company has adopted a standard size varying from 300 to 550 H.P. They are particularly well suited for replacing smaller steam plants of old vessels.

This smaller type are direct reversible. The reversing gear is based on special patents and is of a different design from that adapted for the cross-head main engines, previously described, being specially suitable for the smaller trunk engines.

The smaller, reversible type of engines are always of the 6 cylinder construction; this number of cylinders gives the engine perfect balance so necessary in marine work in particular. These engines are particularly suitable for installation in the stern of the vessel, thus obtaining the advantage of a clear and unobstructed hold.

The engines can be used for single-screw as well as for twin-screw vessels.

With this new design of marine Diesel engines there can under all circumstances be attained an excellent propulsion, but it is also necessary to build a special type, as the marine Diesel engines designed for twin-screw vessels will not yield good results in a single screw vessel, which is quite analogous with experience gained with steamers.

STANDARD MARINE DIESEL ENGINES FOR SINGLE-SCREW SHIPS

(Burmeister & Wain Types)

Type	Number of Cylinders for Engine	Revolutions Per Min.	I. H. P. Normal	Corresponding I. H. P. for Steamers
6 x 125	6	84	750	680
6 x 150	6	84	950	850
6 x 200	6	82	1150	1030
6 x 250	6	77	1350	1210
6 x 275	6	75	1600	1400
6 x 300	6	72	1900	1650
6 x 400	6	70	2300	2000

Long Stroke Cross Head Engines for Ocean Going Single-Screw Cargo Ships Adapted to a Speed of 9 to 12 Knots.

THE WINTON MARINE DIESEL ENGINE

The principal features of the Winton Diesels are: The use of an enclosed crankcase, trunk pistons instead of the usual cross-head arrangement, and crankshaft bolted up to its bearings, which are mounted in the upper half of the crankcase.

The engines, which are of the four-cycle construction, are produced in the following three sizes:

Six cylinder, 11x14 inches, known as Model W35;

Six cylinder, 13x18 inches, known as Model W24A;

Eight cylinder, 13x18 inches, known as Model W40.

Air Compressor: The fuel is forced into the cylinder, against the compression, by a pressure averaging about 850 to 900 pounds per square inch. The compressor furnishing the air is located on the forward end of the engine. It is of the three stage construction. The compressor piston is of the trunk type and is operated by a connecting rod, which is a duplicate of the connecting rod in the power cylinders and a single-throw, counter-weighted crankshaft which is bolted to the main shaft, the throw being somewhat less than that of the power shaft. Following each stage of compression the air is water-cooled, so that on delivery it is at normal temperature.

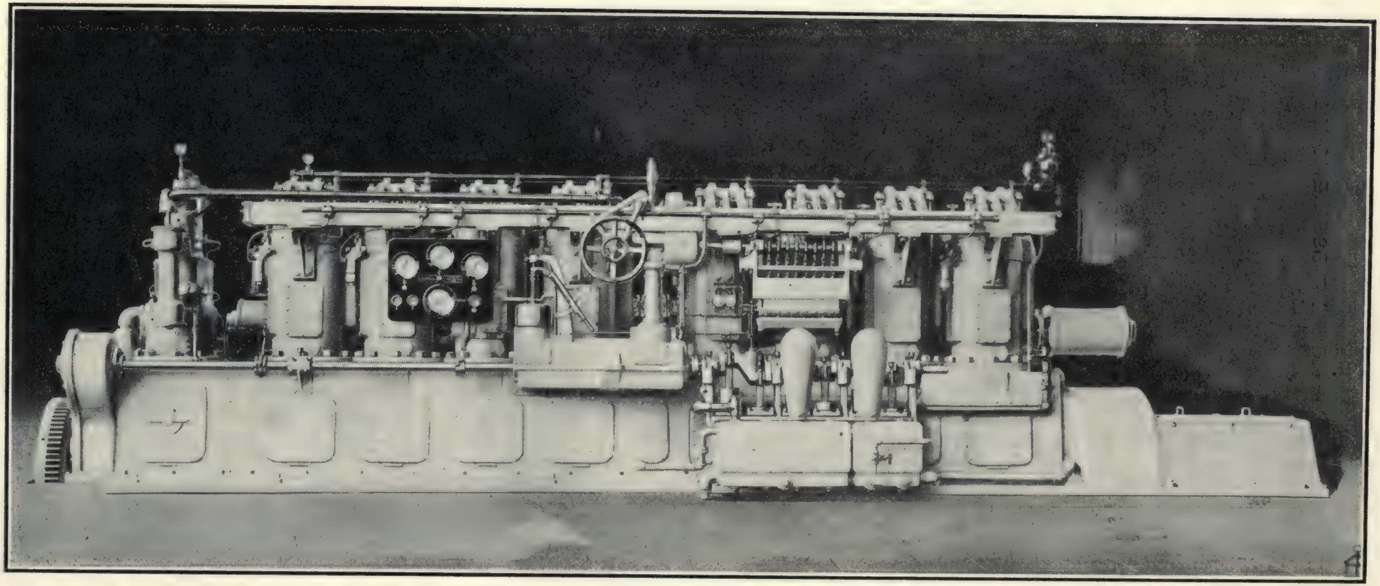
The capacity of the air compressor is considerably in excess of that required for the injection of the fuel and provides for the initial charging and maintenance of pressure in the storage tanks, or air bottles, which are used for starting the engine.

There are two sets of these air bottles, one carrying the air at about 600 pounds pressure per square inch, which is admitted to the cylinders in turn, to force the pistons down, starting the engine rotating. The second set of air bottles carries air at 1,000 to 1,200 pounds per square inch. This is used to inject the fuel into the cylinders. As soon as fuel is injected, combustion takes place, performing the cycle of operation.

Injection Operation: The valve timing conforms to costumary practice. The governor is of the fly-ball type, and acts upon the fuel cut-off valve which meters out the exact amount of fuel to the fuel pump that is required by the load the engine is carrying.

At the time the injection valve is opened, the amount of fuel that has been delivered by the fuel pump is blasted into the cylinder by the high pressure air.

Valve Arrangements: The cam shaft is supported in a long, box-shaped casting, open at the top, which in turn is supported on brackets from cylinders. This trough is partly filled with oil, insuring ample lubrication to the cams. Driving gears and cam shaft bearings are fed directly with oil under pressure from pump.



A Winton "Model W-40." This type of engine may be found on numerous yachts, specially built for long voyages. It is an excellent machine also for ships of largest capacities, suitable for twin and single installation.

The cam shaft is driven by bevel gears from a vertical shaft near the center of the engine, which in turn is driven from a lay shaft and a train of spur gears at front end.

The cam shaft is provided with two complete sets of cams, for forward and reverse motion. Shifting the shaft endways by means of a hand wheel and suitable mechanism, brings either set of cams into operation. The shift from full speed ahead to full speed stern is frequently accomplished in 5 to 6 seconds. Considering the weights of parts which must be brought to rest and started in motion in the opposite direction, this is an exceptional performance.

Each pair of inlet and exhaust valves is operated from the cam shaft by a single rocker arm, the valves being connected by T-head slide on which the end of the rocker rests. Slide is carried in contact with the rocker at all times by a coil spring in center of its hollow stem. Each end of the slide carries an adjusting screw and lock-nut, by which the clearance between the ends of the valve stem and slide can be adjusted. This clearance is 0.015 inches for both main inlet and exhaust valves. This pair of valves are used instead of single large valves, because small valves are much less apt to warp.

The fuel injection valve is located in the center of head and is also operated by a rocker from the cam shaft. The adjusting screw for this valve is set to leave 0.010 in. clearance when the valve is closed. All these clearances are for a cold engine, and will be somewhat more as the engine warms up.

OVERALL DIMENSIONS AND WEIGHTS OF WINTON DIESELS

Model	Weight	Length		Width		Height		Depth below base	
	lbs.	ft.	in.	ft.	in.	ft.	in.	ft.	in.
52	11,000	9	0	3	8	5	2⅛	1	2¼
53	16,000	10	4¼	3	8	5	2⅛	1	2¼
54	22,000	13	4½	3	8	5	2⅛	1	2¼
58	30,000	13	9½	5	1	6	6	2	4
W35	44,000	18	2	5	1	6	6	2	4
W24A	66,000	24	1	5	10¾	7	0¾	2	9¼
W40	90,000	28	10	5	10¾	7	0¾	2	9¼

Model	Bore Inches	Stroke Inches	Power	No. Cylinders
52	7½	11	50 H.P.	3 Cylinders—Reverse Gear
53	7½	11	75 H.P.	4 Cylinders—Reverse Gear
54	7½	11	115 H.P.	6 Cylinders—Reverse Gear
58	11	14	150 H.P.	4 Cylinders—Reverse Gear
W35	11	14	225 H.P.	6 Cylinders—Reverse Gear
W24A	12$\frac{15}{16}$	18	300 H.P.	6 Cylinders—Reverse Gear
W40	12$\frac{15}{16}$	18	450 H.P.	8 Cylinders—Reverse Gear

NOBEL DIESEL MARINE ENGINES

In following detail information of the Nobel Diesel of the latest design, outstanding feats and features may be summed up as follows:

Rating, Speed and Weight: With a rated capacity of 1,600 brake horsepower or 2,000 indicated horsepower at a speed of 106 revolutions per minute, the engine weighs, including a 13-ton flywheel, scavenging pumps, air compressors, etc., but 170 tons, or 236 lbs. per B.H.P.

Overall Dimensions: The total length of the engine is 7.8 meters or 25 ft. 7 in.; its height above the center of the shaft to the top of the fuel valve is 5.9 m., or 19 ft. 4 in.

Fuel Consumption: At full load the fuel consumption of the engine is 0.395 lb. of oil per B.H.P. hour with a heating value of 9,960 calories per kg., or 17,900 B.T.U. per lb. Comparing well in this figure with the highest record established by four-cycle engines. At lighter loads the fuel consumption of this engine is considerably less than that of any four-cycle engine of equal proportion.

Mean Indicated Pressure: At the rated load, the M.I.P. is 6.48 at 92 lbs.

Mechanical Efficiency: At full load the mechanical efficiency is 81 per cent. At 10 per cent overload the mechanical efficiency is 82.3 per cent.

Overload Capacity: The heaviest overload so far carried is 22 per cent, the engine developing 1,958 B.H.P. at 108 R.P.M. continuously during a period of three-fourths of an hour, without any serious complication.

General Information: The engine follows the two-cycle principle. The arrangement of its scavenging ports in the cylinder walls and its scavenging pumps, are valve controlled, with the exception of the scavenging air. Inasmuch as the scavenging ports possess a greater height than the exhaust ports, a greater efficiency in supercharging is accomplished.

The engine is of the open "A" frame type, bolted to the bedplate as well as to the cylinders. Commendable features are the one-sided cross-head slippers, the links and rocker arms which serve to actuate the scavenging pumps, the air compressors and circulating pumps, built to follow the general prevailing marine practice.

The engine has four working cylinders, each resting upon the two cast iron columns composing one "A" frame. Rocker arms, connected to the cross-heads, drive in the following order, the combined low and high pressure stages of the injection air compressor, the two scavenging pumps and the intermediate stage of the air compressor. All this machinery is supported by brackets fastened to the bedplate directly opposite to the frames. This arrangement has the advantage by which it utilizes the available space in the best manner, particulalry in the longitudinal direction of the ship, which is of the greatest value, assisting in estab-

lishing accessibility and a more satisfactory arrangement in pump operation.

The scavenging air is pumped into the hollow frames of the engine which serve as air receiver. The frames are connected to each other by means of distance pieces of sufficiently large cross section.

The operating platform is arranged near the top of the engine, where all the vital parts for operating and maneuvering are within easy reach of the operator. If it should be preferred to have the operators stand on the main floor, no doubt this could be accomplished, but would require some additional and more or less complicated gearing.

Cylinders and Port Arrangement: The inner liner of the cylinder is made of close-grained cast iron and with a mild shrink-fit inserted into the outer jacket. The parts of the cylinder surrounding the exhaust ports are provided with drilled vertical holes in order to secure a most effective water cooling. The exhaust pipe, connecting all four cylinders, is also water cooled. An extension to the cylinder on the opposite side carries the piston valve which prevents the exhaust gases entering here and which controls the scavenging air. This valve is actuated by a push-rod from a cam on the layshaft above. The scavenging air enters from below, leaving the hollow frame casting which has previously been mentioned, serves as air receiver.

In the center of the cylinder head is the fuel valve of standard design, on one side the air-starting valve is arranged; on the other side is the compression relief valve, which is combined with the safety-valve, and which relieves the compression in order to facilitate starting. The air which during starting is compressed in the working cylinders, is not permitted to go to waste, but escapes into the air receiver and intermingles with the scavenging air.

It may be mentioned here that this arrangement in adding to the scavenging air, is very valuable, especially since it takes place in the beginning of the operation, before the scavenging pumps have been able to fill the receiver with air of the required pressure. It eliminates any trouble which may occur in starting the engine.

The pressure of the scavenging air is kept exceedingly low; at full load and full speed it only amounts to 1.6 lbs. per square inch. All these factors contribute to reduce the work expended for the scavenging pumps to a minimum. At full load and normal speed it amounts to about 3.5 per cent of the total indicated horsepower of the engine. The advantage of low pressure, in dispensing with artificial cooling, is easily understood. By accurate test it was found that the rise of temperature at full load only amounted to 10-12° C. or 18-21° Fahr.

The compression in the cylinder is at slow speed 38.5 lbs., and the pressure in the scavenging air receiver but 0.21 lb. The injection air pressure at this speed is 430 lbs., in fact, only slightly higher than the compression in the cylinder. It is here mentioned that even at normal speed the compression is 430 lbs., the scavenging pressure 1.6 lbs., and the injection air pressure 860 lbs.

Operating and Reversing Features: In maneuvering we may summarize the operating performance in following: In bringing the eccentric shafts in neutral position, the fuel valves are put out of motion and the puel pumps are cut out, thereby the engine is brought to a standstill after a few revolutions.

By turning the reversing lever, by means of a gear segment and a rack, the camshifting rod is moved, which in its turn brings the cams for the reverse rotation into their power position. Simultaneously, by a system of levers and rods, the scavenging air controlling valves are brought into their correct position for the reverse, whereupon the starting can be effected in the usual way.

The various levers are mechanically interlocked, so that it is impossible to move the reversing lever and shifting rod, unless both eccentric turning levers are in their neutral position and on the other hand none of these levers can be moved, unless the reversing levers and with it the other parts, cams and valves are in their proper position either for forward or astern running.

In starting the engine a careful investigation should be made that all piping is properly filled with fuel and the required amount of air is in the compressed air tank.

The rollers of the starting valve and of the relief valve are brought into contact with their corresponding cams and at least one of the starting valves will immediately open and will admit air to its cylinder. The air pressure sets the engine into motion, and after one or two revolutions, as soon as the engine has received sufficient momentum, the engineer throws the turning levers into the normal running position, which sets the starting and the relief valve idle and puts the fuel valve into service.

Main Dimensions

Working Cylinders:	
Diameter	26.574 in.
Stroke	36.200 in.
Scavenging Air Pumps, Double Acting:	
Diameter	36¼ in.
Stroke	26¾ in.
Diameter of Plunger Guide	7⅞ in.
Air Compressor:	
Diameter for the three stages,	22¼ in., 9⅞ in., 4 5/16 in.
Stroke for all three stages	22½ in.
Water Pumps, Single Acting:	
2 Pumps for cooling cylinders, etc.	
Diameter x Stroke	5 in. x 14 in.
2 Pumps for general service, bilge, etc.	
Diameter x Stroke	5 in. x 14 in.
Circulating Pump for pistons	
Diameter x Stroke	5 in. x 6¼ in.

Crank Pin:

Diameter x Length ------------------------------------ 15¾ in. x 19 in.

Main Bearings:

Diameter x Length of Journal -------------------- 15¾ in. x 27¼ in.

THE VICKERS DIESEL ENGINE

There is no doubt that in the past many failures in Marine Diesel Engines have been due to the lack of sea-going marine knowledge among those responsible for the design and manufacture of the machinery. Difficulties which had been overcome in the best reciprocating steam engine were often unwittingly resurrected in the early Diesel design, and new mechanical troubles were invited. In surmounting these the engine-room staffs were compelled to spend much of the time they would otherwise have devoted to mastering the purely Diesel features of the engines.

Heavy oil engine work is not a new line with Vickers. They, practically alone of British makers, have evolved an engine of their own, which since 1909 has been accepted as the British Standard for their submarine service, for which it was designed. In the few cases in which foreign designs have been tried against it in similar circumstances, Vickers engines have proven their equal.

The engine referred to is of the high duty express type, of which over 380,000 B.H.P. are made or in hand, and the experience gained with this class of engine renders it a comparatively easy task to develop the slow running type of marine engine required for mercantile work. In addition, they have made two pairs of 750 B.H.P. engines running at 150 R.P.M. which have given satisfactory service.

A brief summary of the Vickers engine now specially designed for mercantile use is given in the following pages:

The engine is of the four-stroke cycle, this type having been adopted by the Vickers Company. The first point of departure from all other Diesels is in the adoption of a fuel injection system in which the air compressor is entirely done away with. For some years now this system has been adopted by Vickers on all their engines to the entire exclusion of the air injection system, notwithstanding that Vickers are manufacturers of air compressors and are therefore fully alive to the improvements in modern machinery. Nevertheless their opinion, which will probably be shared by most experienced sea-going Diesel engineers, is that in the compressed air system lies the main source of trouble and danger in the modern marine Diesel engine. For this reason, although saving of weight and space is not so important in merchant ships as in submarines, for instance, they recommend solid injection for mercantile work. Another reason is that the system is very foolproof and lends itself to successful operation by a comparatively unskilled personnel and in case of necessity can be kept running when an air injection engine,

owing to variations of adjustment, would be dangerous to start. There is nothing mysterious in the system, in fact, its simplicity is a revelation to many. A proof of its efficiency is found in the fact that the Vickers air spraying Diesels supplied prior to the development of the Vickers injection system were subsequently converted at the owners request. Contrary to uninformed comment from some quarters, this system gives full consumption results better than many spraying systems, consumptions per hour down to 0.378 lbs. per B.H.P. having been obtained in the official tests of Vickers engines. This would correspond to about 0.28 lb. per indicated horsepower in an ordinary air injection engine.

The disposition and number of the auxiliary pumps on the main engine depends on the arrangement of the engine room. The standard design permits of swaybeams being fitted to the two end cross-heads from which the required pumps may be driven.

EFFICIENCY ACCOMPLISHMENT BY M. S. "NARRAGANSETT," EQUIPPED WITH VICKERS ENGINE

1st February 1921.

M. V. "NARRAGANSETT."

	long spells of bad weather	Very bad weather for a few days				Starboard propeller damaged	
						Very bad weather during passage	
From	New Orleans	Barrow	New York	Liverpool	New York	Manchester	New York
To	Liverpool	New York	London	New York	Liverpool	New York	Birkenhead
Date from	4.7.20.	25.9.20	16.10.20.	16.11.20.	4.12.20.	25.12.20.	14.1.21.
To	22.7.20.	9.10.20.	30.11.20.	30.11.20.	17.12.20.	10.1.21.	25.1.21.
Distance, miles	4,528	3,013	3,204	2,985	3,028	2,987	3,030
Running Time, hours	402	325½	313	317	289	377	287½
Average Speed, actual, knots (Trial trip 10.5, designed)	11.1	9.25	10.2	9.41	10.48	7.92	10.53
Mean Revolutions of Engines (118 designed)	118	111	117.3	114.4	117.9	111.42	116.7
Eng. Fuel used per day, tons	9.61	9.36	10.51	9.23	10.76	9.56	10.41
Boiler Fuel used per day, tons	2.13	2.16	1.71	2.95	2.03	3.37	1.87
Total Fuel Consumption per day, tons	11.74	11.52	12.22	12.18	12.79	12.93	12.28

According to information received from owners regarding similar oil-fired steam vessels, the average speed is 10.25 knots, and the daily consumption of oil works out at 35 tons, while the daily consumption of fuel in a coal-fired steam tanker very slightly larger than the "Narragansett" is 55 tons at 9.8 knots.

NOTES: During the winter runs the engines had often to be eased owing to the bad weather necessitating a reduction in the speed of the ship. The vessel returned to Barrow after July, 1920, for opening out and examination of machinery, and for uncompleted items of work to be made good, no opening out having taken place after trials.

Details of "Vickers" Solid Injection Engines

One of the problems in connection with the development of the marine oil engine in motorships which is attracting the greatest attention among engineers concerned with this branch of engineering is the efficacy of the solid injection system, as compared with the ordinary blast or pressure injection principle which has hitherto been mainly adopted for motors for mercantile service. For this reason, if for no other, the latest types of Vickers engine deserves mention.

As perhaps the fuel system is of the greatest interest from the engineering standpoint, we will first deal with the arrangement adopted, which is novel in many respects. The principle lies in forcing fuel oil at about 4,000 lbs. per square inch direct into the cylinder through a special type of valve in the cylinder cover—a system which has been used by Vickers for many years past on all their submarine, monitor and tanker engines, and has proved its reliability under arduous conditions.

The fuel is supplied under this pressure by means of a small battery of four pumps of the ordinary plunger type. The plungers are driven from four eccentrics operated from the horizontal shaft, which itself is driven by spur gearing from the crankshaft. The fuel from these pumps is taken through a small box to the main pipe line and thence a pipe is tapped off to each cylinder in connection with a small shut-off cock in each case. It would be difficult to devise any simpler arrangement, and the choice of four pumps is merely a matter of convenience, having no direct relationship to the number of cylinders. There is a hand pump close to the main fuel pumps for priming the pipes, with a delivery to the fuel box. The control of pressure of oil fuel is effected through a small lever acting on the suction valve of the pumps in the usual manner through a spindle. This pressure, which can easily be maintained at practically 4,000 lbs. per square inch, can also be varied within smaller limits by another small hand wheel. Its effect is the same as that of the lever.

We will now deal with the valve gear and method of control, which, owing to the singular method employed should be carefully studied.

The camshaft being driven by gearing by means of a sloping shaft, which is itself driven from the crankshaft through bevel gearing, supplied by forced lubrication from the main bearing oil lubricating system at a pressure of about 20 lbs. per square inch. On this camshaft are two side-by-side cams for the operating of each valve—exhaust, fuel, inlet, and starting. These cams are enclosed. No mechanically operated starting valve is needed in the cylinder cover, where there is only a non-return valve—a simplification with some advantages. The exhaust valve, inlet valve, and fuel valve are operated by levers; the levers being mounted on the maneuvering shaft and actuated from the cams on the camshaft.

The method of reversing can now easily be followed, since it is clear that the following operations have to be carried out:

1. The valve levers have to be lifted off the cams on the camshaft.

2. The camshaft has to be moved fore and aft to bring the astern cams underneath the rollers of the valve levers, after which the levers must be dropped down again on the cams.

3. Compressed air has to be admitted to all six cylinders, then two have to be placed on fuel and four on air; next, two on air and four on fuel, and finally all on fuel.

If the engine is running when the order is given to stop, the hand wheel is turned to stop position as indicated on the dial. This causes a partial rotation of the spindle, which raises or lowers the rods. These are attached to sleeves, on which the levers operating the fuel valves are eccentrically mounted. The other end of the lever on the fuel valve cam is, therefore, raised from the cam by this operation and is only brought down on to the cam at the right moment by the movement of the starting wheel. In other words, when the engine is in the stop position the fuel valves and starting air valves are automatically out of operation until the hand wheel is moved.

Assuming the engine is stopped after having been running ahead, and the order is received to go astern, the reversing lever is moved from the back position to the front. This puts compressed air on the Servo motor, which by means of a rack motion, first partially rotates the horizontal shaft—which lifts the exhaust and inlet valve levers off their cams through the link, then causes the lever to move fore and aft, giving the corresponding motion to the camshaft, after which, by the continued rotation of the shaft and the movement of the link, the valve levers are once more brought down on to the cams. Only when this complete movement has been effected is it possible to move the starting wheel.

Immediately the cams are in the astern position this starting wheel is rotated by hand until the indicator on the dial shows that air being supplied to all six cylinders through the distributing valves behind this wheel. There are three of these valves with three main pipes, each leading to two of the starting valves. The engine then starts up on air, after which, with an almost imperceptible pause, the starting wheel is turned to the next position indicated on the dial, namely two cylinders on fuel and four on air. This is accomplished by the rotation of the spindle as previously mentioned, allowing two of the fuel valve levers to come down on their cams. Further rotation of the starting wheel cuts off the air supply and allows four of the six cylinders, and finally all of them to operate on fuel.

It should be mentioned that the valve levers are lifted off their cams by the movement of the maneuvering shaft, owing to the fact that these levers are mounted eccentrically upon the shaft. The reason that the fuel valve levers are brought down on to their cams in pairs as described is that there are cams on the shaft which lift the levers at the time required for putting into action the respective valves, according to the position of the starting wheels.

It may be noticed that there is a hand pump, operated by a lever, in case it is desired to carry out the reversal by hand instead of by compressed air, in which case the small lever is pulled over to the forward or hand position.

We are not debating in this article the advisability of either four-stroke cycle or two-stroke cycle as found preferable by the Bush-Sulzer Engineers, Nordberg's, etc., but it is but fair to add here that the Vickers Ltd., claims a matter-of-fact figure on economy in fuel consumption in their adopted four-stroke cycle engines, which by official test of British officials is reduced to 0.378 lb. per B.H.P., corresponding to 0.278 lb. per I.H.P., in an ordinary air injection engine. This they claim is principally accounted for through their method of injection system.

SIZES, WEIGHTS AND DIMENSIONS OF "WESTERN" DIESEL ENGINES

"Western" Diesel Engines now are manufactured in standard size of 25 B.H.P. per working cylinder, in multiples up to six cylinders. With following data obtainable, sizes, weights and dimensions are as follows:

No. Cylinders	Brake H.P.	R. P. M.	Shipping Weight Domestic	Shipping Weight Export
1	25	325	10,000 lbs.	12,000 lbs.
2	50	325	14,000 lbs.	15,750 lbs.
3	75	325	17,500 lbs.	19,250 lbs.
4	100	325	20,000 lbs.	24,000 lbs.
6	150	325	28,000 lbs.	32,500 lbs.

Shipping weights given are for standard commercial engines.

In the "Western" Diesel engine the atomized fuel is injected into the highly compressed air in the cylinder, igniting on its own compression similar to all Diesel process.

In the open type nozzle fuel injection the fuel first enters in a passage-way which opens into the main combustion chamber of the engine, and, immediately the oil has been deposited in this passage-way, a blast of highly compressed air from an outside source drives it through small openings with such force that it enters the main combustion chamber in such a highly atomized state that complete combustion takes place as soon as it mixes with the compressed air contained therein.

DESCRIPTION OF THE 600 B.H.P. MARINE WERKSPOOR ENGINES INSTALLED IN THE TWIN-SCREW MOTOR TANKER "CHARLIE WATSON" (STANDARD OIL)

Total Brake Horsepower 1200.

The engines are of the four-cycle, single-acting marine crosshead type, and each develops 800 indicated horsepower, or 600 brake horsepower at 165 revolutions per minute. The cylinder blocks, connected by an intermediate piece over the maneuvering station, carry each a set of three cylinders. The cylinder block at the same time forms a water jacket around the cylinders, and is supported by cast-iron frames on the crosshead guide block and by the opposite guide block columns which are bolted to same. The cylinder block and the guide block are again supported by lower columns, which rest on and are bolted to the bedplate. Heavy steel tie rods run from the top of the cylinder block through the bottom of the bedplate, with nuts on each end.

With this design, each set of three cylinders has one combined crosshead guide block, which is supported by lower columns. After lifting four tie-rods on one side of the engine, these small columns are taken out and the crankshaft can then be removed.

The built-up crankshaft is 10⅝ inches in diameter. The main bearings are of the square box type, the square boxes carrying the bottom brasses, which are made so that they can be rotated and come clear.

The piston rods pass through stuffing boxes of the drip pans which collect carbon and used oil from the cylinders and prevent same leaking into the lower oil-tight part of the engine, which is provided with forced lubrication. The lubricating oil enters through the binder of each main bearing and is forced through the crankshaft to the crankpin and through the hollow connecting rod to the crosshead. A drip pan bolted under and to the bedplate collects the return oil, which is again pumped by an electrically driven pump through strainers back to the main bearings.

The cylinder and cylinderhead are cast in one piece. It provides an efficient water-cooling around the cylinder top, and furthermore increases the cooling water circulation between the valve housings.

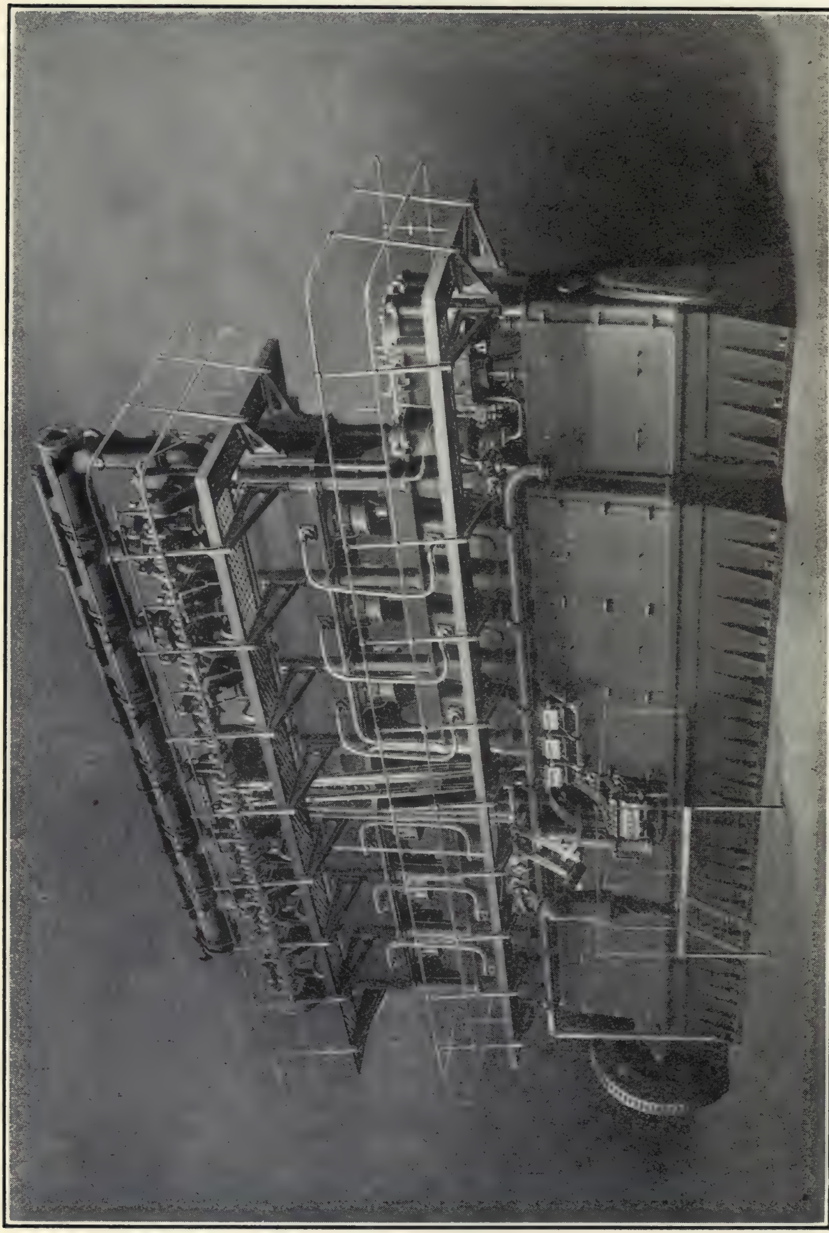
In casting the cylinder and head in one piece, allowance has been made for easy examination or removal of the piston from below.

This is done by placing the crank on the bottom center and dropping the cylinder extension or skirt onto the drip pan, which then exposes the full length and top of the piston and allows for easy removal of the rings. By the loosening of four bolts on the end of piston rod, the piston can be lifted down.

The piston is air cooled and provided with eight rings.

The intake and exhaust valves are of cast iron, with steel stems.

The fuel needle can be easily removed and ground in place while the engine is running.



One of the Two 850 B.H.P. Werkspoor Diesel Engines for the Motor Tanker, H. T. Harper. Built by the Pacific Diesel Engine Co.

This also applies to the safety valve, which at the same time is used as relief valve, its spring being relieved while the engine is reversed.

All valve springs are outside the valve cage, in order to keep them cool and free for inspection.

The engine is fitted with the new Werkspoor reversing gear. The rockers are mounted on skew eccentrics, keyed to the reversing shaft, which, when turned over 180 degrees, lift and transfer the rocker rollers from the ahead to the astern cams, and vice versa.

The camshaft is actuated by four connecting rods from the half-time shaft below, which is driven by a spur gear from the main crankshaft. The gear wheel on the lower-half-time shaft contains the governor, which acts on the fuel pump when the engine speeds up over 165 revolutions per minute.

The lower half-time shaft carries two eccentrics, each operating a set of three plungers of the fuel pump, which is bolted on the bedplate at the maneuvering stand. The fuel pump has to take care of an equal fuel distribution to all fuel needles, and therefore has a separate plunger for each cylinder.

Each plunger delivers fuel only when its suction valve is closed. The suction valves are lifted by two rockers which are actuated by the crossheads of the pump and which pivot around an accentric. The pivoting points of these rockers are lowered or raised by turning the eccentric through either the hand levers on the maneuvering stand or through the action of the governor. When lowered, the suction valves close earlier and the pump delivers more fuel; when raised, the suction valves close later and the pump delivers less fuel. Three adjusting screws in each of said rockers, one under each suction valve, allow an accurate adjustment of the amount of fuel to each cylinder. A pyrometer in each exhaust enables the engineer to check the amount of fuel fed to each cylinder in comparison with the others.

The fuel from the fuel pump passes through a manifold which is placed on top of the frame between the two cylinder blocks. This manifold contains a second delivery valve for each fuel plunger, and is called the cut-out block, as it carries two cut-out valves for the injection air, one to each set of three cylinders. These two valves are operated by the hand levers from the maneuvering station.

There are also six spindle valves for the purpose of cutting out, by hand, the injection air to each cylinder separately if ever found necessary.

The air compressor is provided with crossheads and is driven by a crankshaft which is bolted to the forward end of the main crankshaft.

The compressor piston rods run through stuffing boxes of a diaphragm drip-pan. The cylinder liners and the three corresponding coolers are set in one cast-iron block, which forms at the same time a water jacket. This arrangement allows the pistons and coolers to be easily removed from the top.

Besides delivering air to the injection air-bottle, the air compressor charges the starting air bottles from the intermediate pressure cooler. The air inlet to the low pressure cylinder can be regulated by hand. All valves are of the plain disc type.

The engine is reversed from the maneuvering station by an air ram, cushioned by an oil cylinder and connected to a vertical shaft fitted with a gear rack which rotates the camshaft 180 degrees to the ahead or astern position.

At the maneuvering station are two hand levers, each of which controls at the same time the starting air and fuel supply to the engine.

On the right of the maneuvering station are the forced feed lubricators for the main cylinders and those of the air compressor, while directly on the left are pressure gauges for lubricating oil, low pressure air, intermediate pressure air, starting air, and injection air. On the bedplate at the maneuvering station are mounted the high pressure fuel pump and a fuel hand pump, all placed at a central point from which the engineer can watch the action of the engines.

While the Werkspoor engine has been patterned after their original design, nevertheless for use in the United States considerable departures from European mechanical arrangements are considered advisable by American manufacturers. As is universal with four-cycle marine engines, the valves are actuated by cams on the horizontal camshaft.

Inasmuch as the Werkspoor engines are adhering to cross-head construction, hence the piston rod is short, the cylinders are supported by vertical steel cylindrical columns. The inclined cast-iron columns being mainly for the purpose of taking the thrust due to the connecting rod.

Following the usual procedure of four-cycle engines, the arrangements of the valves are in similarity to the Burmeister & Wain, McIntosh & Seymour, etc., there being four in the cover of each cylinder. The fuel inlet valve, being located in the center is notable for its difference from valves of this kind of other types of machinery in Diesel construction. Where springs are used to hold the same in its seat, a lever attachment serves to hold the same securely in its place, the valve in this case being held down by a spring on one side exerting its pressure at one end and acting against the force of the cam, the lever assisting in its functioning. A valve of this kind may easily be replaced and at all times can be quickly examined.

The reversing method of the Werkspoor is exceedingly simple. Similar to the four-cycle large types of engines its procedure is carried out. The maneuvering levers are raised clear of the cams, the consequential shifting of the cam shafts back and forward accomplishing the reversing, which is followed after this with bringing the levers back in desired position.

WERKSPOOR MARINE ENGINES.

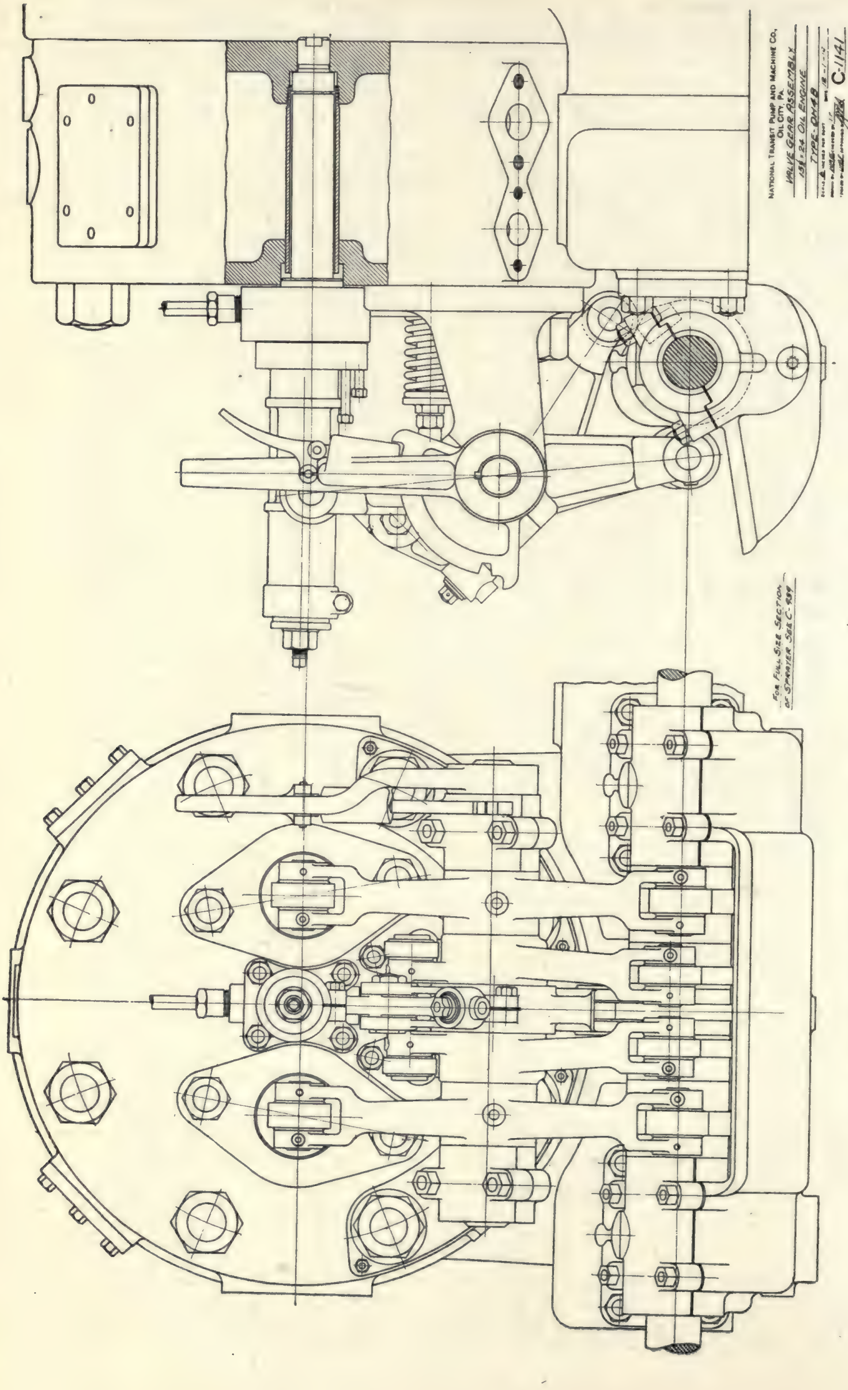
B. H. P.	100	150	200	300	450	600	850	1100	1500	2000	3000
I. H. P.	135	200	267	400	600	800	1140	1470	2000	2670	4000
R. P. M.	275	225	225	225	200	165	135	115	100	90	90
No. Cyl.	3	3	4	6	6	6	6	6	6	6	8
Length	9′ 10″	11′ 2″	13′ 2″	18′ 2″	25′ 0″	31′ 9″	38′ 3″	42′ 6″	47′ 6″	52′ 6″	57′ 0″
Weight	14	16	20	31	60	115	163	200	250	330	480
Fuel 24-hr. bbl.	3¼	5	7	10	15	20	27	35	48	65	95

*Engines up to and including 450 B.H.P. are trunk piston type; all engines above this size are crosshead engines.

WERKSPOOR STATIONARY ENGINES.

B. H. P.	65	130	200	265	350	525	750	1100	1500	2000	2750	4000
R. P. M.	300	300	300	300	250	250	180	150	130	115	105	100
No. Cyl.	1	2	3	4	4	4	6	6	6	6	6	8
Length	6′ 0″	8′ 0″	10′ 0″	12′ 0″	15′ 0″	19′ 0″	30′ 0″	36′ 0″	40′ 0″	45′ 0″	50′ 0″	54′ 0″
Weight	10	14	18	22	45	65	120	170	210	250	345	495
Fuel 24-hr. bbl.	2.1	4.25	6.5	9	11.5	17	25	36	49	65	89	130

*Engines up to and including 525 B.H.P. are trunk piston type; all engines above this size are crosshead engines.

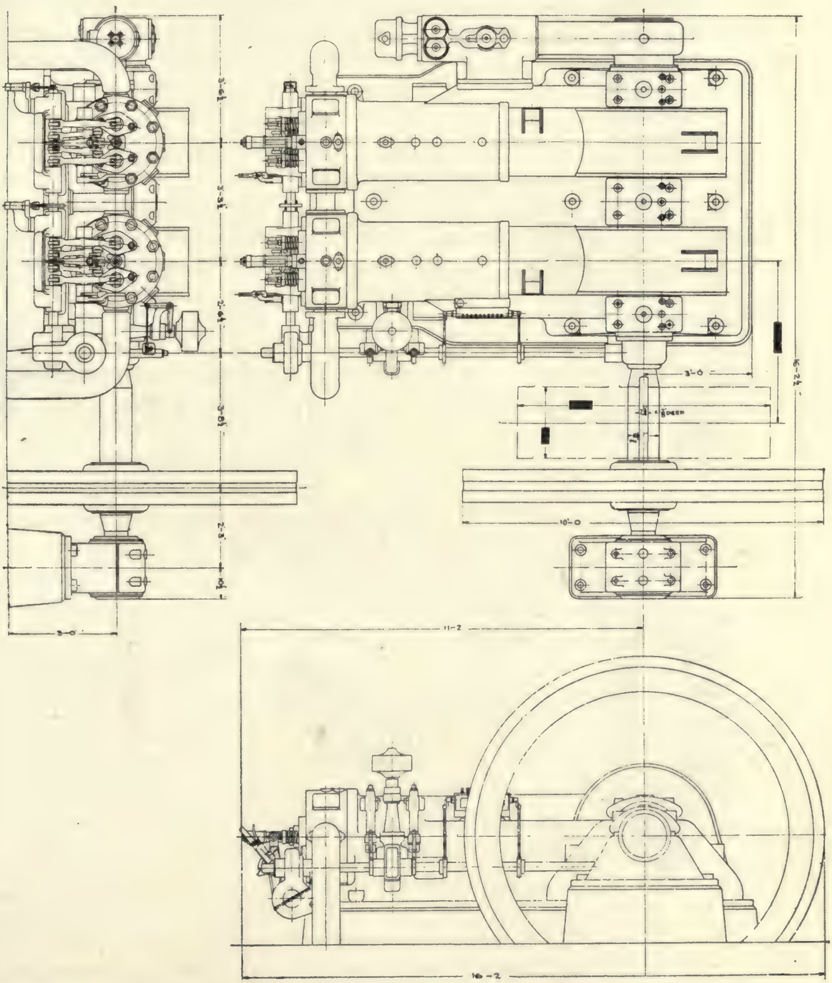


Valve Gear Assembly of 15½ × 24 National Transit Oil Engine. Type DH 4B.

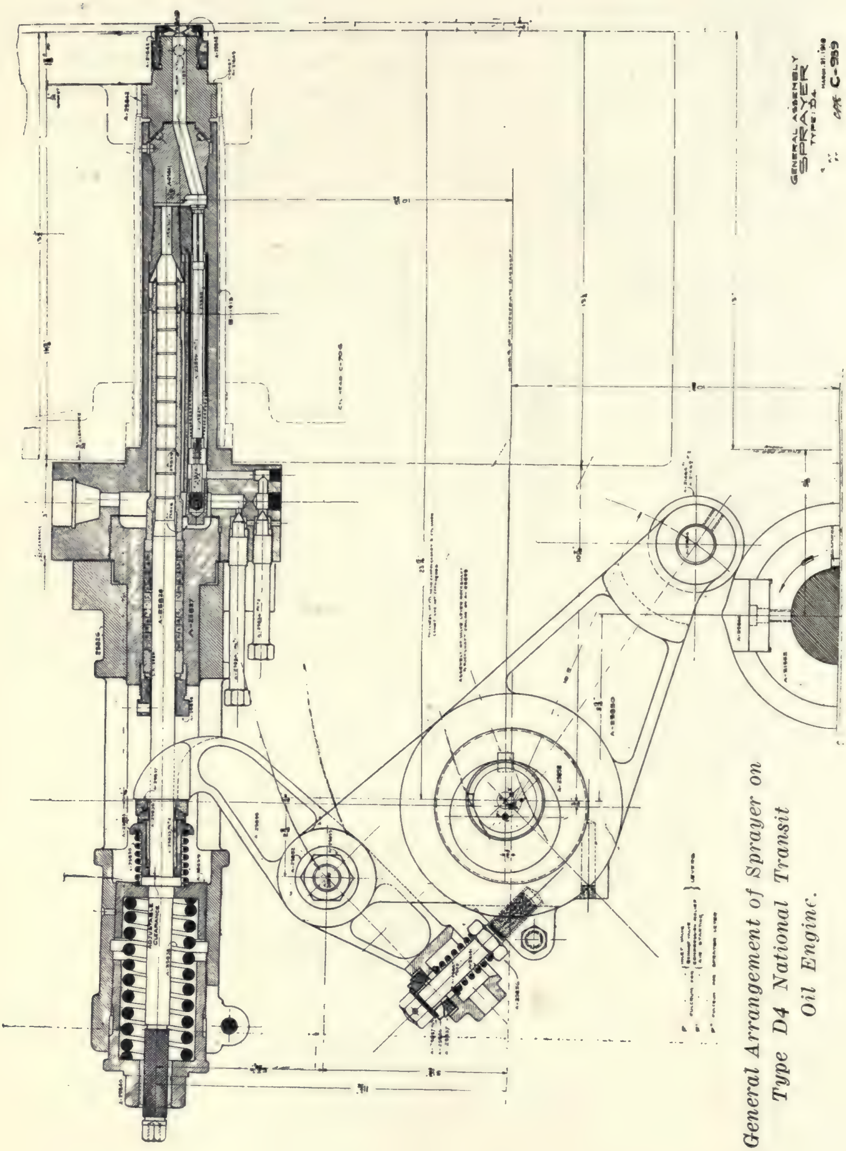
NATIONAL TRANSIT DIESEL ENGINES

The National Transit engines have novel departures from the types of older construction. Features, such as advanced pump equipments, modern valve arrangement of exclusive National Transit design, latest compressor construction, etc., are well deserving of highest comment.

As will be observed by the accompanying illustrations of the National Transit types of twin engines, as well as double engines, the machines are suitable as factors where the requirements call for stationary power producers. The company has made a study of Diesel engine design corresponding with modern features of exclusive American principle. Satis-



Plan View showing detail arrangement of 15½ × 24 Twin Oil Engine.



General Arrangement of Sprayer on Type D4 National Transit Oil Engine.

factory results dealing with economy establishments have been noted, principally by adhering to established uniform construction of original designs with added late improvements.

The illustration of the National Transit Twin Oil Engine, of size 15½x24 inches, shows the constructive advantages of this particular classification. Both cylinders and the bedplate are one casting, a significant feature eliminating undue stresses and assuring rigid construction. Each cylinder is provided with a liner.

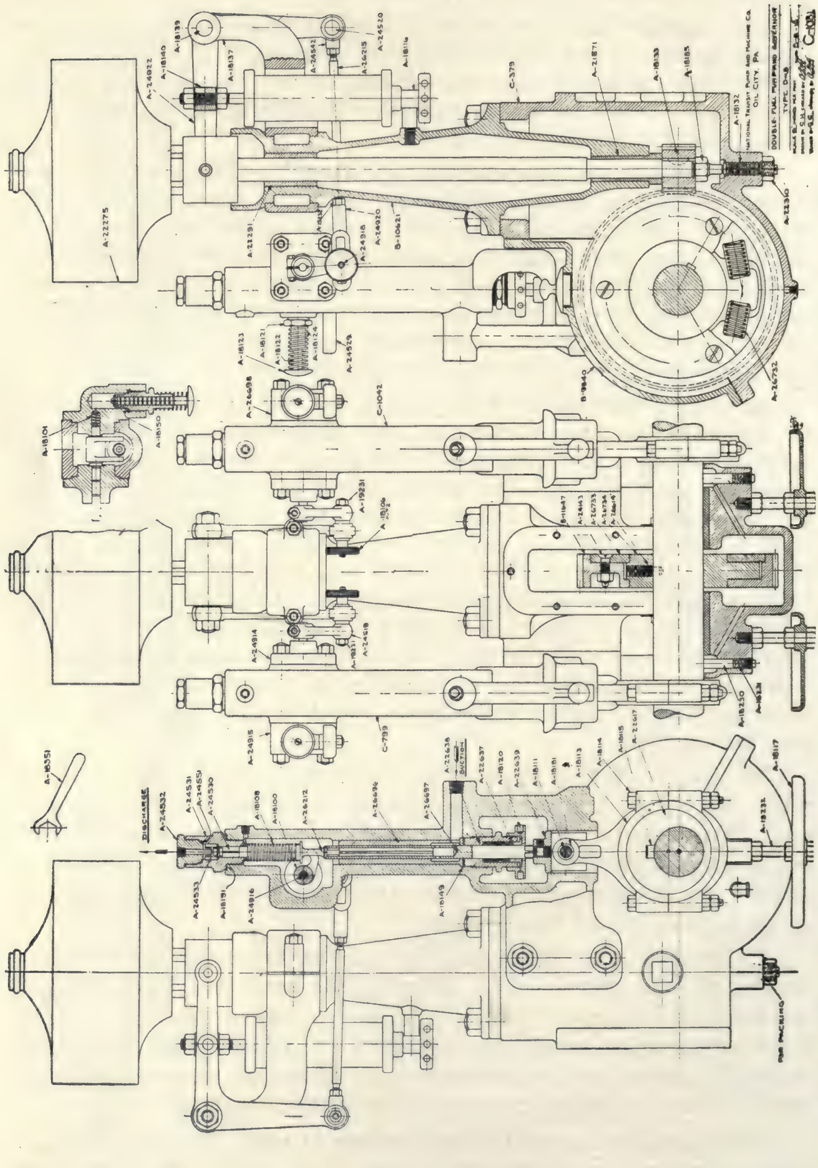
Three main frame **bearings** and an outboard bearing are shown supporting the flywheel and the crankshaft. Sufficient space is left between the flywheel and main frame to allow the installation of a belt pulley. The lay shaft is driven from the crankshaft by means of cut spiral gears. It serves to drive the valve gear, governor, sprayer, air starter and lubricating and fuel oil pumps. In this respect it should be noted, that the general arrangement allows accessibility to all parts for inspection or adjustment when the requirement calls for the same.

As will be observed from the illustration that the **main fuel pumps** and **governor** are mounted together in a single assembly on the right hand side of the main frame. Each fuel pump is driven by an eccentric on the lay shaft, and each side of the engine has a separate fuel pump. The pump plungers are of the differential type, the upper or large part being hollow and having a cut-off valve seated at its upper end. Each cut-off valve is under direct control of a Jahn's governor. Each plunger has a full positive stroke with each revolution of the lay shaft. Governing is effected by allowing the cut-off valve to seat at a pre-determined point in the upper-stroke of the plunger, thus delivering to the sprayer a quantity of oil correctly proportioned to the load which the engine is carrying. The governor, as previously stated, is driven by spiral gears from the lay shaft.

Spraying is effected by the open nozzle type, and located in the extreme out end of the combustion chambers. Fuel oil from each fuel pump is deposited in the open channel of each sprayer body. As soon as the valve controlling the compressed injection air opens, this oil is injected into the combustion chamber thoroughly atomized, and is ignited by the process of heat temperature.

The problem of suitable material for **cylinder heads** has been considered on this engine. All cylinder heads are made from specially selected cast iron with large cored water passages. The large water jacket arrangement in no way endangers the thermal efficiency of the engine, but rather acts in harmony with temperature normalizing necessary in reliable operation of internal combustion machinery.

The **intake** and **exhaust valves** operate in cages, either of which is easily withdrawn by removing two nuts. The valves are operated by cams through rocker arms. The cams are mounted on a shaft, in front of and below the cylinder head. This shaft is driven from the lay shaft by bevel gears.



Double Fuel Pump and Governor. Type D3 National Transit Diesel Engine. (Twin Engine).

A single two-stage compressor for spraying the liquid fuel into the power cylinders is bolted to the engine. This compressor is driven by a connecting rod from the main crankshaft of the engine. The piston is of the differential type. The cylinders and also the valves are amply water-jacketed. An inter-receiver, which is virtually an inter-cooler, is provided between the high and low stage air compressor cylinders. The high stage air is discharged at about 1000 lbs. pressure into a seamless steel discharge pipe which acts as a receiver and which conveys it to the sprayer.

THE NORTH BRITISH DIESEL ENGINE

This type of engine is of the four-cycle construction. Its eight cylinders cast en bloc are of 26½ inches diameter, with 47 inches piston stroke.

The shaft horsepower of the engine is rated at 2,000 B.H.P., or 250 B.H.P. per cylinder, and the piston speed at 96 R.P.M. is 817 feet per minute.

The engine develops normally at 96 R.P.M., 2,380 indicated horsepower, or 292 I.H.P. per cylinder, equivalent to steam driven machinery where twin engine motor power is installed, to 4,500 I.H.P.

In connection with this it should be noted that the engine is rated at the very moderate mean-effective pressure (on B.H.P. basis) of approximately 80 lbs. per square inch.

In considering the foregoing figures it should also be remembered that the fuel injection air compressors are not driven from the main engines, but are in twin sets in duplicate driven by auxiliary Diesel engines of the same type as the main engines, but with trunk pistons and run at 250 R.P.M., whereas the main engines are of the single acting cross-head type.

All pistons on main engines are internally water cooled, a telescopic pipe and jet system being employed.

Its reversing is accomplished by a system of camshaft and eccentric, assisted by levers causing the camshaft to fall and raise in its respective operating position.

THE STEINBECKER DIESEL ENGINE

(Note.—This engine is a late development of exclusive German design. Its remarkable adaptability to utilize Tar-Oil for fuel purposes, with its consequential special design to accomplish the burning of the same, is herewith set forth.)

The engine of the future probably must be able to operate continuously on coal tar-oils as well as asphaltic oils. While the problems to

be solved in the design, manufacture and operation of such engines are now greater than those connected with engines utilizing lighter fuel oils, experience will solve them; the world's supply of the more volatile oil fuels is not inexhaustible and sooner or later the heavier fuels must be quite commonly used.

One of the German firms devoting much attention to the development of engines burning very heavy oils, such as tar-oils, is Friedrich Krupp, Germaniawerft, Kiel-Garden, Germany, who in addition to building their own cross-head and trunk-piston types of Diesel engines, have a license for the manufacture of the Steinbecker engine. The first 100 H.P. two-cylinder engine has been developed under the personal supervision of the inventor, Mr. Steinbecker.

The Steinbecker engine has no compressor and might be called a combination of the surface-ignition and full-Diesel principle.

Principle of Operation: The principle of operation is as follows: Towards the end of the compression stroke the fuel pump forces a small quantity of oil through the horizontal-channel into the vertical-channel, the top end of which is fitted with a bulb with a number of spray holes, the bottom end being open to the cylinder. As the air rushes from the cylinder into the bulb it atomizes the oil in the same manner as water is atomized in a flower-spray as used by florists, and the mixture of air and oil is carried into the bulb. When the piston reaches the top of the stroke this mixture of oil and air is ignited by the heat produced by compression, resulting in great increase in pressure and a back-rush of the burnt gases which carry into the cylinder the oil fuel which the pump has meanwhile pumped into the vertical channel. In the cylinder the mixture burns and expands in the same manner as in other Diesel engines.

Special Features: It will thus be seen that this Steinbecker engine is a Diesel engine without a compressor, which atomizes the fuel-oil by blowing it with great velocity into the combustion chamber by means of gases which are formed by exploding a small amount of fuel-oil in a hot retort.

This engine is claimed to be less complicated and therefore cheaper to build than the usual full-Diesel type; the fuel-needle-valve, injection air-bottle, air compressor, and high-pressure air piping are eliminated. For starting the engine from cold a small auxiliary sprayer is provided, which may be put out of action when the engine is running.

THE WORTHINGTON SOLID INJECTION DIESEL ENGINE

An Advanced Method, Burning Fuel Oil in Small and Medium Sizes.

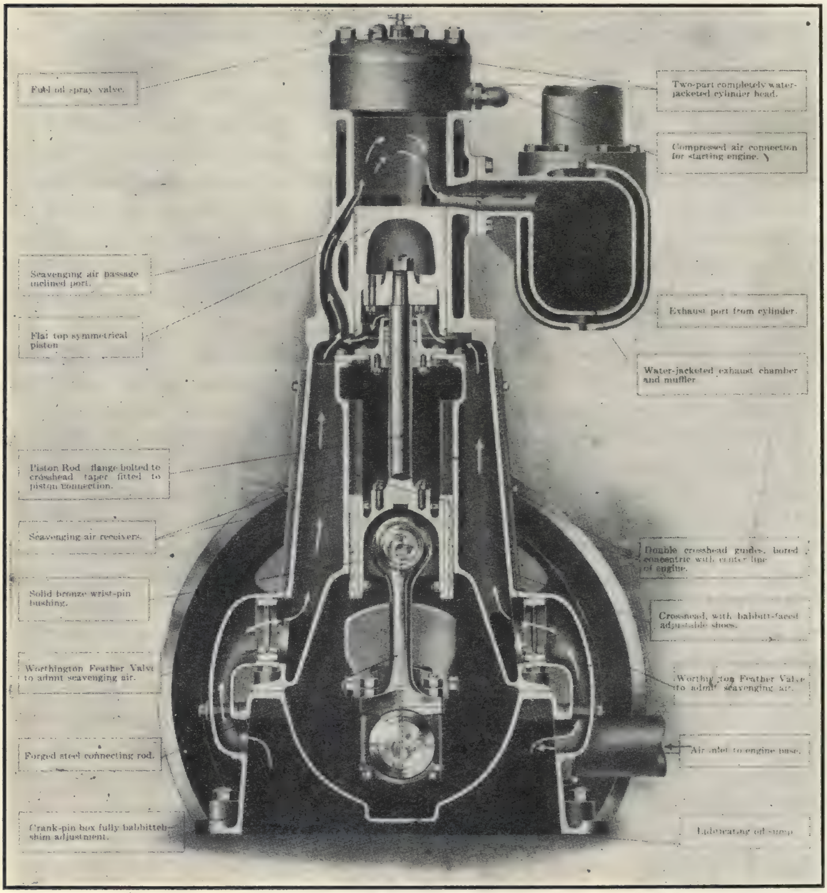
The Worthington Solid Injection Engine has a new form of combustion chamber, inherently controlling the combustion time and rate, independent of time of pump injection. This new Worthington solid injection Diesel engine has no operative limitations of size, is capable of burning all fuel oils of the air injection engines, and has all of the elements of simplicity and reliability so necessary in practical operation.

Worthington Divided Combustion Chamber, Injection Chamber, Ejection Orifice and Cylinder: The special fuel burning and combustion control feature, is a divided combustion chamber, wholly water jacketed as in Standard Diesel engines, but differing from them in having two parts connected by a fuel ejection orifice. One part of the combustion chamber, that between the cylinder head and the top of the piston, holds about three quarters of the air. The other part, which is the injection chamber, holds about one-quarter of the air at the end of compression. Combustion takes place in two stages, and starts with injection, the first oil entering being ignited by the hot air.

Injection Chamber—Combustion Limited: Fuel is injected by the pump as a spray directly into the injection chamber, where the full charge of oil could not meet more than one-quarter of the air, even if all the air in the injection chamber came into contact with all the oil. Partly by design of the spray nozzle to give a suitable form to the spray, and partly by the shape of the injection chamber, the oil is prevented from coming into contact with more than a part of the injection chamber air, so that even less than one-quarter of the total is active in burning oil during injection. As a consequence the pressure cannot rise very much during injection, no matter how fast the injection nor how much too early the pump may be timed, within reasonable limits. If only half of the injection chamber air is active, then not more than one-eighth of the oil could burn during injection; if one-quarter of the air came in contact with the oil, not more than one-sixteenth of the oil charge could burn and the pressure could rise only one-sixteenth as much as if the oil were suddenly sprayed into the whole air charge of an air injection Diesel engine combustion chamber.

Main Cylinder Combustion Automatically Graduated by Pressures on Ejection Orifices: After injection of the oil into the injection chamber with limited air contact, and partial or pre-combustion, the unburned oil has been gasified by the partial combustion, and this oil gas is suspended out of contact with the cold walls in the bottom of the injection chamber, and close to the ejection orifice. It is ready for ejection into the main air charge in the cylinder. The gasified unburned oil, which includes the bulk of the charge delivered by the pump will be forcibly ejected into the cylinder through the ejection orifice at the right time by the outward movement of the piston, which causes the pressure to fall in the cylinder, aided by the slight rise of pressure in the injection chamber, due to the

limited pre-combustion. As it emerges from the injection chamber, the hot gasified oil, accompanied by some unused air from the injection chamber and followed by the rest of it, burns in the main air charge in the cylinder as fast as it flows. By the high flow velocity of the gases passing through the orifice, a violent mixing action is set up by the jet entering the cylinder, that contributes to good combustion.



Transverse Sectional Assembly, Outside Air Passages, Worthington Diesel Engine, Two-Cycle, Solid Injection.

Two-Cycle Cross-head Construction: The air charging of the cylinder is done by the two-cycle method as the best arrangement for the ranges of sizes adopted. In this respect the standard practice in surface ignition engines has been followed, but in the details of carrying out this plan the trunk piston with crankcase scavenging chamber has not been ac-

cepted. Instead, the cross-head construction of the large Diesel motorship engine has been adopted, with the crank end of the cylinder used as a scavenging pump. This arrangement keeps all scavenging air out of the crankcase, permitting the use of circulating forced feed lubrication of all bearings without loss of oil, and by the stuffing box separating cylinder from crankcase, also preventing contamination of main lubricating oil by foul cylinder oil.

Difficulty in Design of Solid Injection Diesel Engine: Nothing would seem to be easier than to attach the solid injection pump and spray nozzle of the surface ignition engine to a Diesel engine, to make a solid injection Diesel engine. This has been tried by many and each learned the same lesson. The combination is hopelessly bad, worse by far than either of the originals. The combustion is bad with heavy smoke and much internal carbon, and fuel consumption is high. Explosive shocks and thumps, or loss of power, or both, are also present if the combination works at all, with possible loss of control by the governor, and in most cases inoperativeness of the injection pumps. Shocks and detonations are due to difficulty of controlling the timing of injection and rate of combustion, which if too early or too fast, always produces this effect, and doubly so if both too early and too fast. Delaying injection will eliminate detonations and explosive shocks, but then combustion will surely be too slow and last too long, resulting in excessive high fuel consumption. Smoke and internal carbon are due to improper sprays, or rather too improper relation of the form and shape of spray to the form of combustion chamber, aggravated by the use of heavier fuel oil than before. Pump and governor control difficulties are due primarily to increase of delivery pressure, but they are increased by the substitution of Diesel grades of fuel oil for the lighter, more fluid grades for which the mechanism of the surface ignition engine was designed.

Development of Solid Injection Principle of Worthington System: Having discovered that the fuel system of pump, governor control and spray valve of the solid injection surface ignition engine, applied to a Diesel engine will not work, the most obvious step is to change the former to fit the latter. This will attain at least one ideal, the elimination of surface ignition hot metal and adherence to the completely water jacketed combustion chamber of the Diesel engine, with high enough compression to ignite fuel by the hot air alone.

Another ideal of greater importance, but also of greater difficulty in attainment, is the prevention of explosive shocks and detonations, by arranging for the non-explosive combustion of the Diesel engine in such a way as to make explosive combustion impossible, and not merely a matter of pump injection timing, which if deranged defeats the aim. Complete commercial success in the solution of this problem was never attained until the numerous experiences through the effort of the Worthington Company have added towards the solution.

Early Solid Injection Diesel Engines—Their Limitations: These two early solid injection Diesel engines are different from each other, and

each has such a limited scope and characteristics as to justify the conclusions that the Worthington solution of the problem, the latest development in the solid injection oil engines, is a real contribution to the small engine field that has so long needed the solution of this problem.

One of these early solid injection Diesel engines is the result of redesign of spray nozzle, fuel oil pump, and controls, for direct solid injection into the ordinary Diesel engine combustion chamber without any change in the latter as to shape or compression carried. Fairly good combustion of fuel oils has been secured in engines of considerable size, but not in small ones commercially, and not without extreme sensitiveness as to adjustment, which must be almost of micrometer exactness to avoid explosive detonation shocks or smoky combustion, or both. In itself this system is so far no more than a demonstration that a Diesel engine can be operated with solid injection spraying, instead of air spraying, and with about the same efficiency. It also proves, however, that there is a need in such cases for some special reliable means of insuring proper accuracy of adjustment of fuel feed and combustion rate, or preferably, of securing automatically the proper control without accurate adjustment.

Small Solid Injection Diesel Engines Without Pump Injection, Fuel Feed Cups: The second of the early solid injection Diesel engines preceding the final solution, is an invention which again uses the standard Diesel combustion chamber, but with a new method of fuel feed, without a timed injection pump. This method of fuel injection successfully prevents the development of explosive shocks by automatic means operated by cylinder pressures that make it impossible for the fuel to enter too soon or too fast. In these engines fuel feed is a two stage operation. During the first stage fuel flows by gravity from a constant level chamber similar to that forming part of any carburetor, past a metering needle valve into a cup projecting into the combustion chamber, and communication with it by very small holes in the side of the cup near the bottom. The cup receives its oil charge before compression starts and the fuel is prevented from flowing into the cylinder by the smallness of the holes at first, and later during compression, the oil is held in the cup by the compressing air that flows through the hole from the cylinder into the cup. No oil can flow out of the cup until the cup pressure is higher than the cylinder pressure, or what is the same thing, until the cylinder pressure falls below the cup pressure, which will surely happen during the first part of the expansion stroke. At this time the oil will escape into the cylinder, accompanied and followed by air from the cup which sprays the oil into the main air that has been compressed to be hot enough to ignite the fuel. This fuel feed cup type of engine, which is a commercial success, again demonstrate the practicability of operating a solid injection Diesel engine, and the importance in so doing of preventing too early or too fast an injection of oil by simple means that does not require sensitive adjustment.

This means of fuel feed control is effective in very small engines, of sizes beginning with one horsepower, competing directly with carburetor

gasolene-kerosene engines, at but little increase in cost and at least equal simplicity and reliability, but it has not been successful in the intermediate sizes up to the smaller air injection Diesel engines, the range of sizes mainly occupied by the surface ignition engines. While it may successfully burn fuel oil, it is ordinarily operated on kerosene to avoid difficulties of flow with various oils. It contributes something toward the solution of this problem, but is not itself a solution any more than the former step noted. Its size limitation is due to the necessity for small holes in the cup to prevent premature escape of fuel into the cylinder, and the difficulty in larger sizes of dividing the oil between many holes, each of which is small enough. Its greatest lesson is the proof that automatic proper control of time and rate of combustion is possible, if special means be devised for the purpose, even though the means used is not of universal application, and that the combustion rate control means may be wholly or partly independent of pump or spray valve timing.

FUEL PUMP AND CONTROL END OF WORTHINGTON TWO-CYCLE ENGINE

Speed regulation is obtained by opening a by-pass and **not by variation** of the length of the fuel pump stroke.

The amount of fuel supplied to the cylinder depends on the time of opening of the by-pass valve. This in turn depends on the angular position of the eccentric shaft, which is controlled by the governor.

The governor, which is located on the end of the engine crankshaft, is connected to the eccentric shaft by suitable links. Any increase in the engine speed from normal will cause the governor to turn eccentric shaft thru a small angle which at the same time will lift end of by-pass lever. When the fuel pump plunger raises the by-pass lever and by-pass plunger, by-pass valve will be opened earlier. As a result of this earlier opening of the by-pass valve, more fuel is by-passed back to the fuel supply reservoir, thus reducing the amount supplied to the cylinder and promptly bringing the speed back to normal, without changing the time when injection starts.

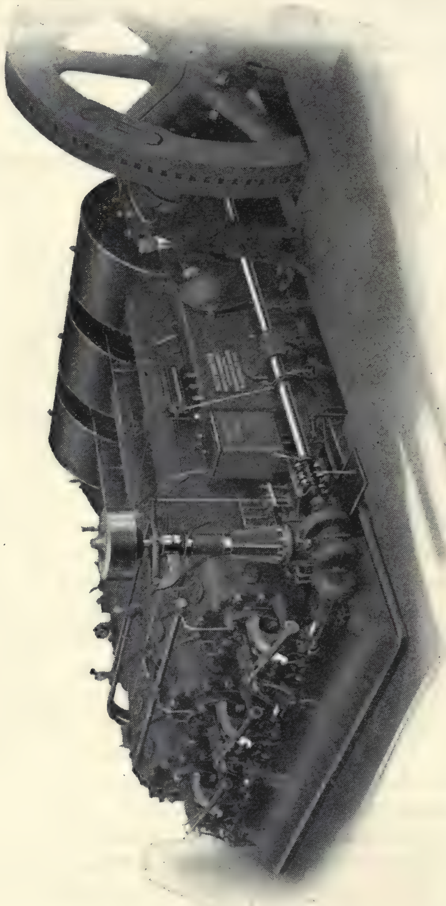
Eccentrics keyed on the engine crankshaft drive the fuel pump plungers through the tappets. The upper ends of the eccentric straps are provided with hardened steel contact rollers and are guided by links, replacing the crosshead and guide construction previously used on older types. The pump plunger tappets pass through a partition which prevents fuel oil leaking down into the control housing and mixing with the lubricating oil. All running parts are splash lubricated by oil from end main bearing, overflowing back to the crankcase sump so as to keep a high level in the control housing.

A hand adjusting screw at the end of the by-pass lever makes it easy to equalize the oil delivery from all plungers on multi-cylinder engines.

WORTHINGTON SNOW OIL ENGINE

The Snow Oil Engine described in following pages, is of the representative type of late advanced development in internal combustion machinery. In it is embodied the best and latest American engineering practice, together with such features of European design as have become standard for this class of machinery.

Unlike the Marine Engine, where the vertical construction is universally adopted and as a matter of fact is advantageous in many respects, the stationary engine appears to be adhering to the horizontal construction.



Over-all View of 600 H.P. Three Cylinder Worthington's Snow Oil Engine With Speed Changing Governor—Fig. 1

For stationary purpose, by the manufacturers of oil engines following the horizontal design, the following advantages are claimed in the horizontal construction in contrast to the vertical method:

(1) There is a better distribution of stresses on the crank shaft and main bearings. (2) Better lubrication of piston. (3) Easier cleaning and repair work, especially to pistons, which can be taken out without removing the cylinder-head or valve gear. (4) Inspection is easier and attendance more convenient. (5) Less height is required in the engine room.

A claim has been often advanced, that in horizontal engines there is a greater wear on the cylinder walls, on account of their carrying the weight of the piston. However, as vertical engines are almost entirely

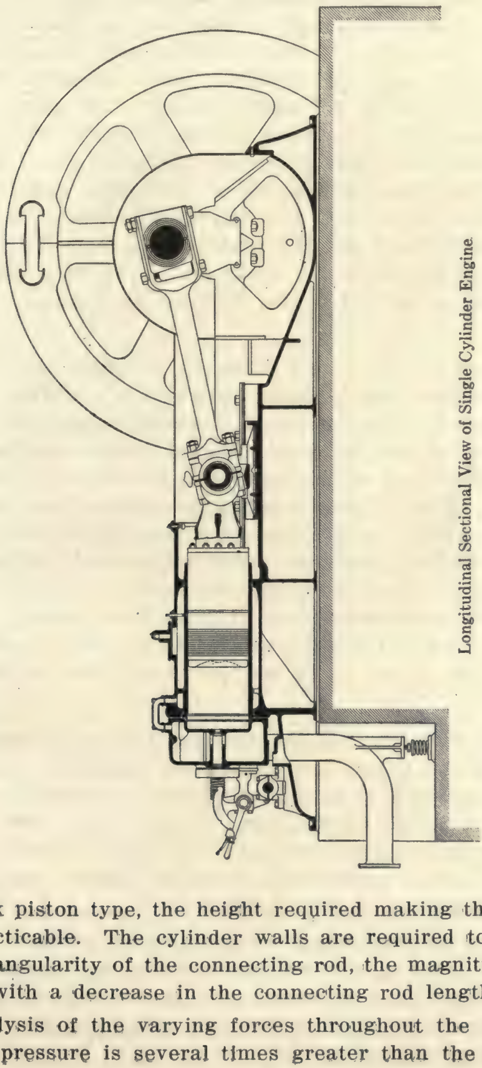


Fig. 2
Longitudinal Sectional View of Single Cylinder Engine

of the trunk piston type, the height required making the use of a cross-head impracticable. The cylinder walls are required to take the thrust due to the angularity of the connecting rod, the magnitude of the thrust increasing with a decrease in the connecting rod length.

An analysis of the varying forces throughout the cycle shows that this thrust pressure is several times greater than the pressure, due to

the weight of a piston carried horizontally. With usual design, increasing the connecting rod one crank length will decrease the thrust by an amount about equal to the weight of one piston. From the foregoing it is apparent that it is of slight importance, whether the piston is carried vertically or horizontally.

The greatest advantage of the horizontal engine lies in its greater accessibility. This applies particularly to engines like the Snow Oil engine, which are built with cross-head. In the case of a vertical trunk piston engine it is necessary to practically dismantle the engine in order to get at the piston or wrist pin bearing. This means that the valve gear and cylinder head must be removed, the connecting rod disconnected at the crank pin box and the piston and connecting rod then lifted out through the upper end of the cylinder. Compare this with the procedure in the case of a horizontal engine. The sheet steel crank splasher is removed, the connecting rod disconnected at the wrist pin end, and the piston then removed through the open end of the cylinder. The actual required time for removal of the piston and cross-head of engine as illustrated in Figure 2 is twenty minutes. The piston and cross-head can be replaced and the engine made ready for operation in thirty minutes more, making the total shut down fifty minutes. The same procedure on a vertical engine requires an average of about ten hours.

The lubrication of the power cylinders in much more easily and effectively accomplished on a horizontal engine. In a vertical engine it is necessary to inject oil into the piston at several points in the circumference of the cylinder bore, while a horizontal cylinder can be thoroughly lubricated by a single feed on the upper side, the oil being distributed by gravity. Further, in all internal combustion engines some of the lubricating oil is carbonized. In the vertical engine this carbon works past the piston rings and falls into the crank pit, where it mixes with the bearing oil. In an engine of horizontal construction, much of this carbon is pushed into the counter bore of the cylinder, from where it is removed by a drain valve. Such of the carbon as does pass by the cross-head of the piston is caught in the frame and prevented from mixing with the bearing oil.

We will give now some of the features adding to economy. The high efficiency of the Diesel type of engine, unequalled by any other method of power production for either marine or stationary purpose, is primarily due to the high compression. It also varies with the degree of thoroughness with which the fuel and air is mixed, so that the economy actually obtained is to a considerable extent dependent upon the design of the parts by which the mixing is effected.

Another feature of considerable importance and peculiar to the Diesel type, is the slight variation in fuel economy through a considerable range of load. This feature is of particular advantage in installations consisting of one unit, and which are required to operate at low load factors for a considerable portion of the time.

It will be interesting here to show established facts of fuel consumption for the Snow Oil Engine:

0.48 lbs. per B.H.P. hour at full load.

0.50 lbs. per B.H.P. hour at three-fourths load.

0.57 lbs. per B.H.P. hour at one-half load.

The guarantees are based on oil having a heat value of 18.500 B.T.U.'s per pound. In operation the fuel consumption is considerably below the guarantees, as shown in Figure 3, which gives the result of a test through wide range of load under ordinary working conditions. This

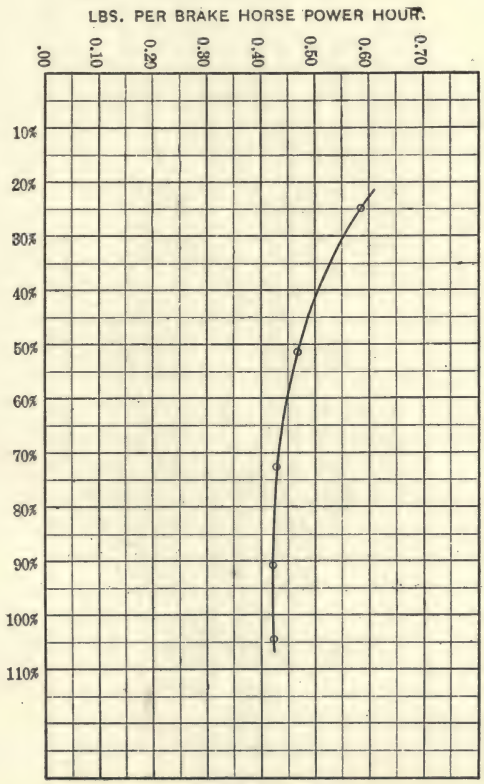


Fig. 3

Fuel Consumption Curve, Four Cycle "Snow" Oil Engine.

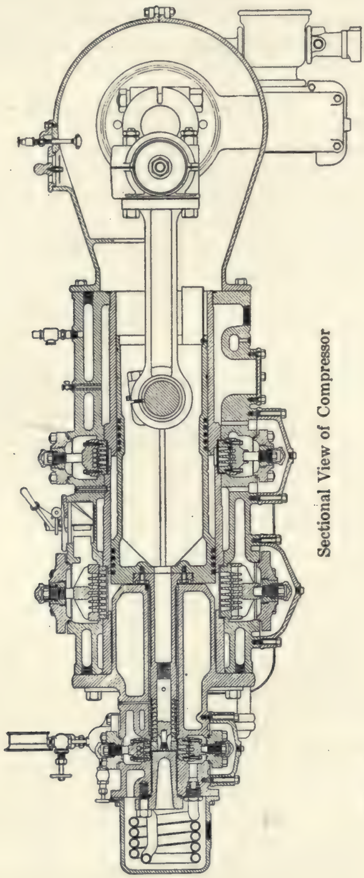
Brake Horse Power—Full Load = 100%

curve shows the fuel economy which can be attained in actual service and also the very slight increase in fuel consumption for a considerable decrease in load factor from full load.

The construction of the compressor, as used on the Snow Oil engine, is shown in the longitudinal sectional view of Figure 4. This compressor is a three-stage type. The cut shows the valves, suction and discharge

for the three stages, both assembled and with the guard and valve strips removed.

As will be observed from the sectional view of the compressor, that it is thoroughly water-jacketed and provided with inter-coolers formed in the jackets for low and intermediate stages. The after-cooler for the high stage air is a pipe-coil placed in the water chamber adjacent to the high pressure cylinder. In addition the discharge valve for all stages are provided with a water jacket of special design, which keeps the valve

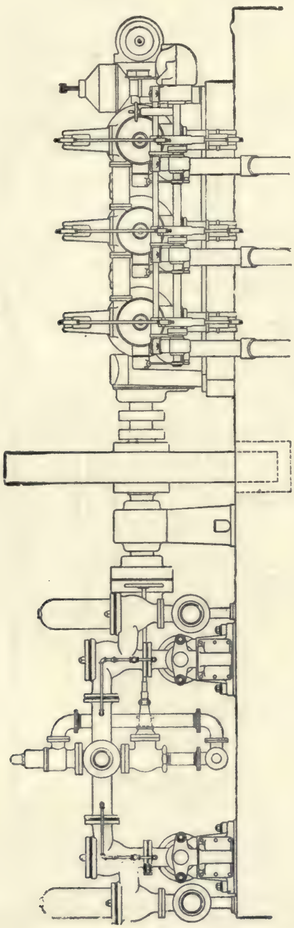


Sectional View of Compressor

Fig. 4

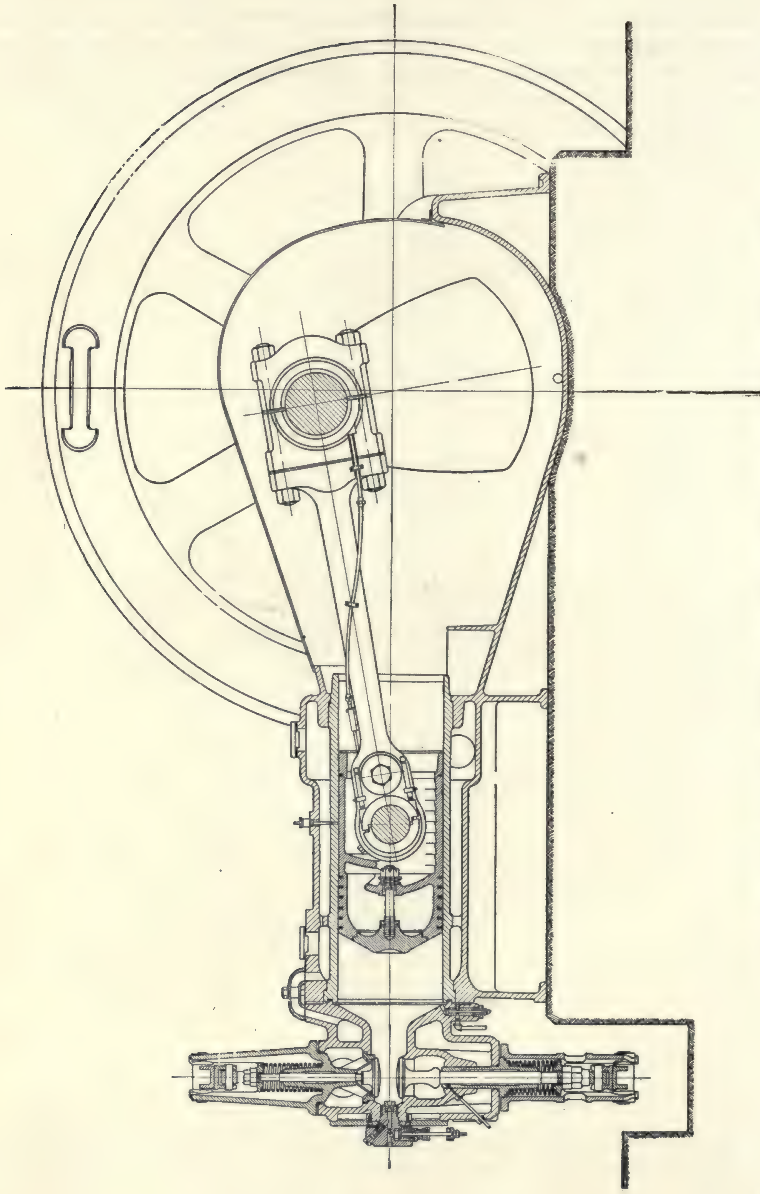
parts at a low temperature, preventing carbonization of the lubricating oil, and assisting the action of the inter-coolers. The efficiency of these cooling arrangements is so great, that the air in the discharge pipe from the high-stage cylinder is cooled sufficiently to allow the hand to be placed on the pipe without discomfort when the engine is in full operation.

Of the different types of spray valves used on Diesel machinery, the Snow Oil engine employs one of a most simple nature. This spray valve is of the open nozzle type, the fuel being deposited in a pocket, which is in communication with the clearance space in the cylinder. When the air valve is opened, the oil is driven by the spraying air through the atomizer from which it issues into the cylinder in a finely divided spray.



General Diagram of Installation of Allis-Chalmers Oil Engines

The open nozzle construction has the advantage that pure, clean air only and not a mixture of oil and air, passes the spray valve, with the result that it is much easier to maintain a perfect valve seat and accordingly to hold the spray air pressure without annoying leakage through valve.



Longitudinal Section Through Allis-Chalmer Oil Engine, Diesel Type

The spray air valve and the fuel oil check valve, which prevents the oil charge from flowing back to the pumps, are contained in a single casing located in the exact center of the cylinder head, thus insuring an even dispersion of the spray throughout the clearance space in the cylinder.

The spray valve and casing can be taken out intact by removing two nuts, or if desired, the spray valve only can be removed for inspection without disturbing the valve casing.

As previously explained, the open fuel injection nozzle is the most important advance toward continuity of service and adaptability for various oils that has been made in the Diesel type of engines.

It consists essentially of an oil receptacle with separate inlets for the oil and air at one end, and connected to the combustion chamber at the other end by a stationary atomizing device. The oil is pumped into the receptacle through check valves during the suction stroke of the engine, and the injection air is admitted through a separate mechanically operated timing valve. Like most modern engines, the Allis-Chalmer arrangement of injection is of the open fuel injection nozzle type.

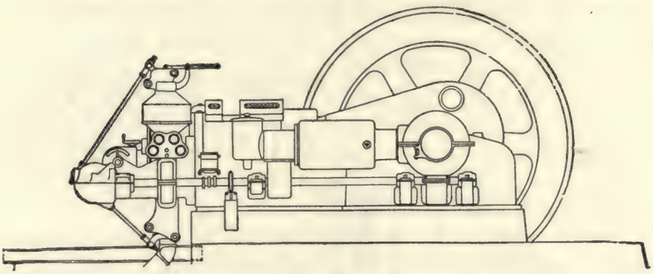


Diagram of Oil Pipe Connection

There is no valve after the oil and air are mixed, thus avoiding cut valve seats. There are no perforated or notched discs with restricted areas and sharp changes in direction to clog with dirt, asphalt or carbonized oil. It does not depend upon the water jackets to prevent carbonizing.

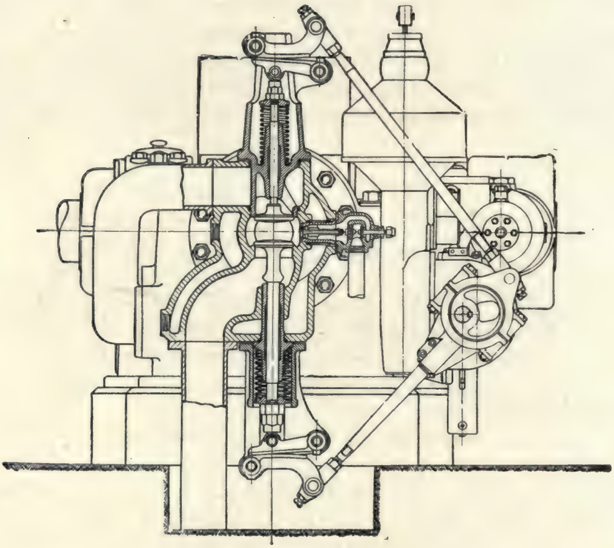
This atomizing is effected by a simple device that does not require a close relation between the size of openings and the amount of oil; so that the maximum power of the engine is limited simply by the amount of oxygen available for combustion and not by the capacity of the nozzle. This gives a remarkable flexibility with swinging loads.

The freedom from clogging permits the use of the lowest grades of fuel without the interruptions of service for cleaning or grinding the valve so frequent with the closed nozzle.

The open nozzle and the mostly adopted scavenging feature of the horizontal type of stationary engines, permits the successful use of any fuel oil that can be pumped, including oils that require pre-heating to make them flow readily through the piping.

Oils such as Texas, Mexican and California natural crudes, some of which cannot be used satisfactorily on steam boilers, are easily handled in the Allis-Chalmers Oil Engine, due to the above stated features. The choice between the fuel oils available is therefore determined solely by the relative cost, heat value, and convenience in handling.

As will be seen by the illustration, the construction is of horizontal design, which permits a simple valve gear arrangement. The motion is transmitted from a single eccentric to the inlet and exhaust valves by rolling contact levers, which has proven by gas engine practice to be the quietest and most durable valve gear known. The use of vertical valves, which are always central with the control seats, insures tight valves and avoids the frequent grinding necessary with horizontal valves,



Valve Gear Diagram, Allis-Chalmers Oil Engine

due to side wear on the stems and the consequent change in valve position in relation to the seat.

The exhaust valve is placed in the bottom of the cylinder head, between the injection nozzle and the cylinder, so that any dirt in the fuel oil will drop out through the exhaust without reaching the lubricated cylinder walls. This location is also favorable for scavenging any dirt out of the cylinder during the exhaust stroke.

The injection air is furnished by a multi-stage compressor, mounted on the side of the main frame and driven directly from a crank on the end of the main shaft. The compressor is equipped with coils for intercooling the air in conjunction with the usual water-stage cooling.

The engine is eliminated from all high pressure bottles in use on most Diesel engines, receiving its starting air direct from the compressor. This separate system of air starting permits the use of pressure from 225 to 250 pounds. The engine is started by opening a throttle valve which admits the starting air from storage tank, to mechanically operated distributing valves, which are entirely separate from the main valve gear. It is not necessary to make any change in the compression or operation of the valve gear in starting.

By referring back to the Snow Oil Engine, we find a similarity existing to that employed on the Allis-Chalmers type. The arrangements for starting the engine are as follows:

An air storage tank of ample capacity is provided and is supplied with air at from 150 to 200 pounds from the spraying compressor on the engine. Each cylinder is fitted with an air-starting valve located in the cylinder head directly below the spray valve, and driven by a cam on same shaft with the inlet and exhaust valve cams. A single lever and quadrant serves to control the air-starting valve, compression relief, and spray valves. With the lever in central and neutral position, both the air-starting valve is engaged and an auxiliary cam is shifted into contact with the exhaust valve lever, by means of which the exhaust valve is held open for a longer period of time and the cylinder compression lowered, reducing the resistance of the engine to rotation. Five or six revolutions of the engine on air is sufficient to increase the air pressure in the pipe between the air compressor and the spray valve to that required for spraying the oil. The governor fuel pump lever is then shifted to place the pump in operation and the main operating lever moved from the inner to the outer position. The movement disengages the air-starting valve and the compression relief cam, and puts the spray-valve in operation, when the engine immediately begins to operate on the fuel and is ready to take the load. The air storage tank can then be recharged by the compressor on the engine.

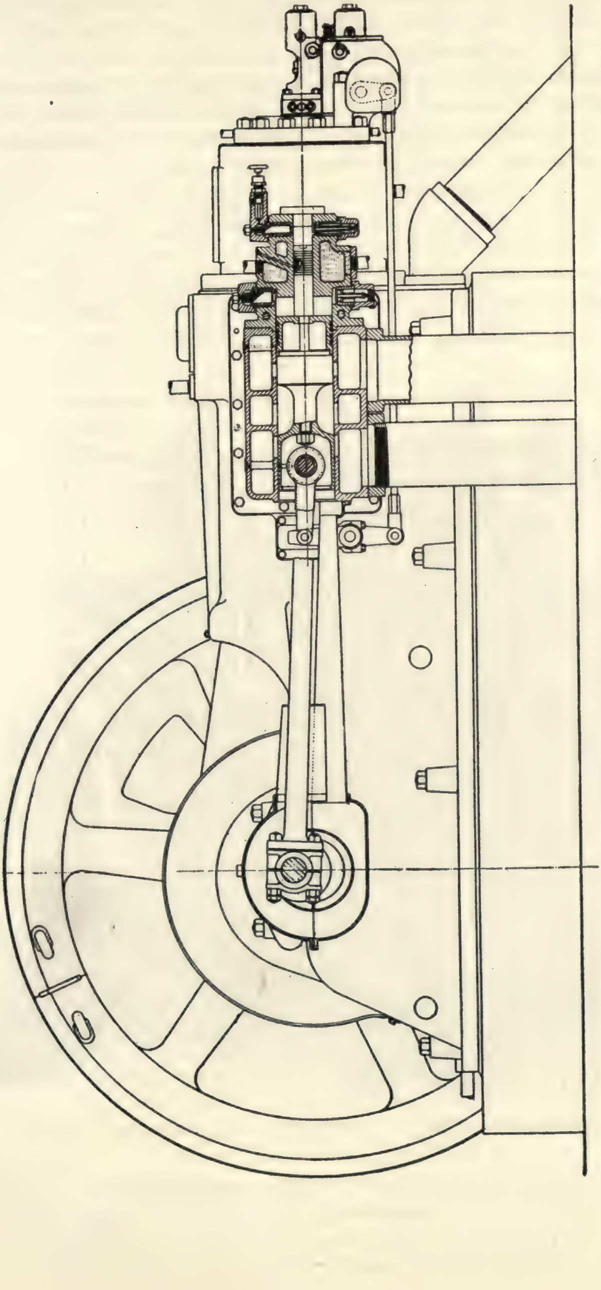
As will be observed on either Allis-Chalmers as well as the Snow Oil Engine, high pressure bottles for the air storage is unnecessary. For the initial start and for emergency purposes afterward, a small independent compressor is used on the Snow Oil Engine, driven by a kerosene or gasoline engine, may be furnished. This is only for use in making the first start and afterward in the event of the air pressure in the storage tank being lost through carelessness or otherwise.

Cooling Water: The amount of cooling water required varies inversely with the difference in temperature of the water as it enters and leaves the cooling jackets. It also varies with the load on the engine. Assuming a 40 degree Fahrenheit rise from suction to discharge temperatures, from 6 to 7 gallons per horsepower hour are required. When fairly pure water is available the discharge temperature may be maintained at 140 degrees Fahrenheit. If, however, the water has a tendency to deposit scale in the jackets, the discharge temperature should not be over 120 degrees Fahrenheit. For this reason, and particularly for hot

climates, where the temperature of the inlet water is high, it is desirable to provide for a cooling water supply of from 10 to 12 gallons per horsepower hour.

For engines without water-cooled pistons, a pressure of from 8 to 10 pounds per square inch is sufficient. For supplying water-cooled pistons the pressure should be from 12 to 15 lbs. per square inch.

Where the water supply is limited, cooling ponds or towers may be used and the same water circulated continuously through the system, so that only a small amount of fresh water will be required to make up the loss due to evaporation.

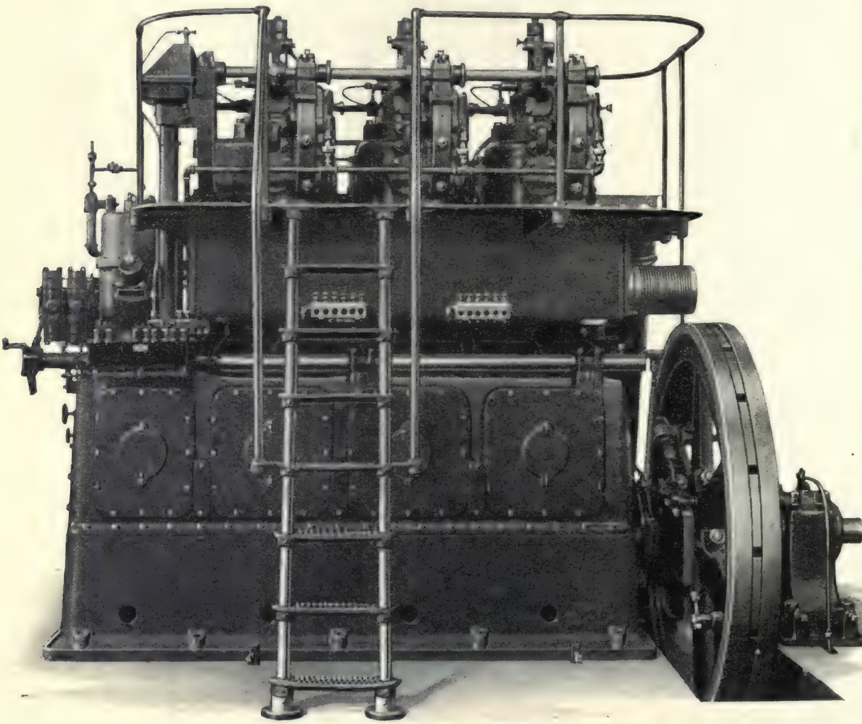


Longitudinal Section Through Working Cylinder of Standard Engine

THE STANDARD FUEL OIL ENGINE

For mechanical simplicity probably the two-cycle crankcase compression engine leads all other types of internal combustion engines. Due, however, to the relatively small amount of scavenging air which can be handled by this system, it being possible to displace not over 60 per cent of the main cylinder volume, and to the restrictions imposed on accessibility by the necessity of a tight crankcase of minimum volume, this type is barred, except for the smaller sizes.

Next in line for mechanical simplicity stands, we believe, the stepped piston two-cycle type employing port scavenging. Moreover, with this type, the restrictions on scavenging air do not exist. In the Standard engine pure air of more than one and one-half times the volume of the main cylinder is forced into it each working stroke. Also, again rather than loss of accessibility is effected, as has already been pointed out.



Exhaust Side of Two-Cylinder Vertical Standard Engine

The adoption of the two-cycle port scavenging principle of operation not only eliminates the complication of inlet and exhaust valves and gear and the time required for inspection and grinding, but also permits the cylinder head to be much smaller and more simple casting.

The fuel pump of the Standard engine is of the variable stroke type and requires for its control no wedges, lever, etc., as it is driven direct by a Rites type of inertia governor, the same as is commonly employed for operating the valve of a simple steam engine.

Also the injection air compressor is a very simple mechanical structure, although meeting all the requirements of a good compressor design. Due to the suction and discharge valves being combined in one unit and in the case of the high stage cylinder, this unit being located in the end of the cylinder bore, there results a cylinder casting with but one valve pocket. Placing of the inter and after cooler coils in the water jacket space of the cylinder eliminates several high pressure joints and also results in a compressor with no hot pipes with which the operator can come in contact, and from which the high pressure air is discharged at but a few degrees above the temperature of the incoming cooling water.

It will be readily admitted that other things being equal, a simple mechanism will naturally be more reliable in operation than one of greater complications. Other features of the Standard Engine which deserve mention here are:

(a) The very ample size of all bearings and other working parts. For instance, with a working cylinder of only 10¾ inches diameter, a main bearing of 6¾ inches diameter is employed.

(b) The conservative manner in which the engine is rated. At full rated load less than 80 pounds M.E.P. is required, as against average Diesel practice of about 100 pounds. This makes for low maximum temperatures and when employed with the large excess of cool scavenging air blown into the cylinder, each stroke results in low mean temperatures during the complete cycle. With normal inlet air temperatures, the exhaust at full load is less than 400 degrees Fahrenheit.

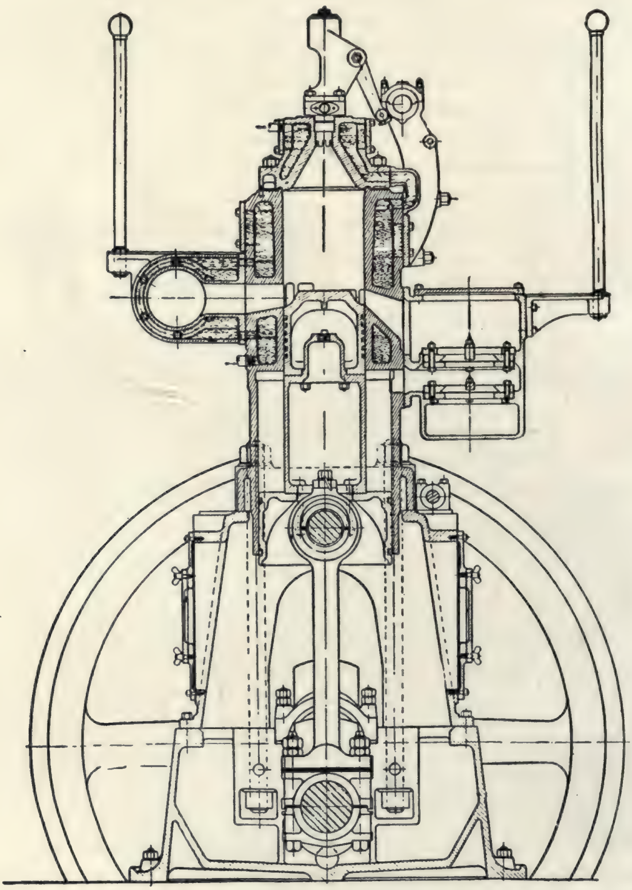
The desirability of low mean effectives in order to secure low mean temperatures is, we believe, sufficient reason for barring devices for obtaining what is known as super compression, even if the extra complications of such devices were not considered.

(c) The employment of the stepped piston which serves not only to furnish a liberal supply of air for cleaning and filling the main cylinder, but which also acts as a cross-head, thus insuring longer life to the main cylinder, since it receives no connection rod thrust. Also the main piston being relieved of the pin and heavy bosses for carrying same becomes an absolutely symmetrical member much better fitted to resist the pressures and temperatures to which it is subjected.

The use of the stepped piston also tends materially to improve crank case conditions and to lower crankcase temperatures as any blow past the main piston will be caught in the scavenging cylinder. The use of a two-piece main piston also serves to the same end by preventing radiation from the hot piston head to the interior of the crankcase. Under all conditions of operation the crankcase will be found clean, cool and free from oil vapor.

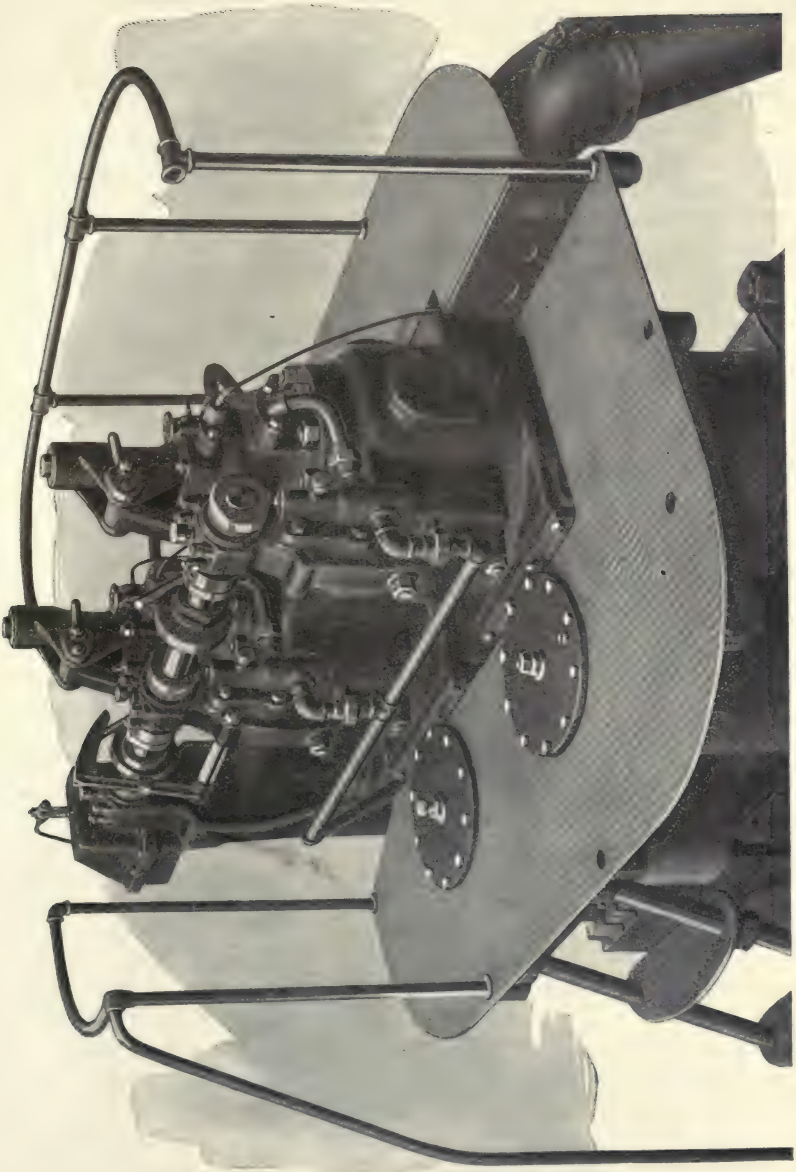
Detail Description—Vertical Type

Box Frame: As previously explained, the Standard engine is of the vertical single acting two-cycle valveless type of Diesel power. A single box frame forms a common support for both the main working cylinders and the injection air compressor cylinder. On account of the through bolts used between cylinder flanges and base plate, this frame is not subjected to heavy tension strains and very liberal openings may therefore be provided for inspection and adjustment of bearings.



Transverse Section Through Main and Scavenging Cylinder

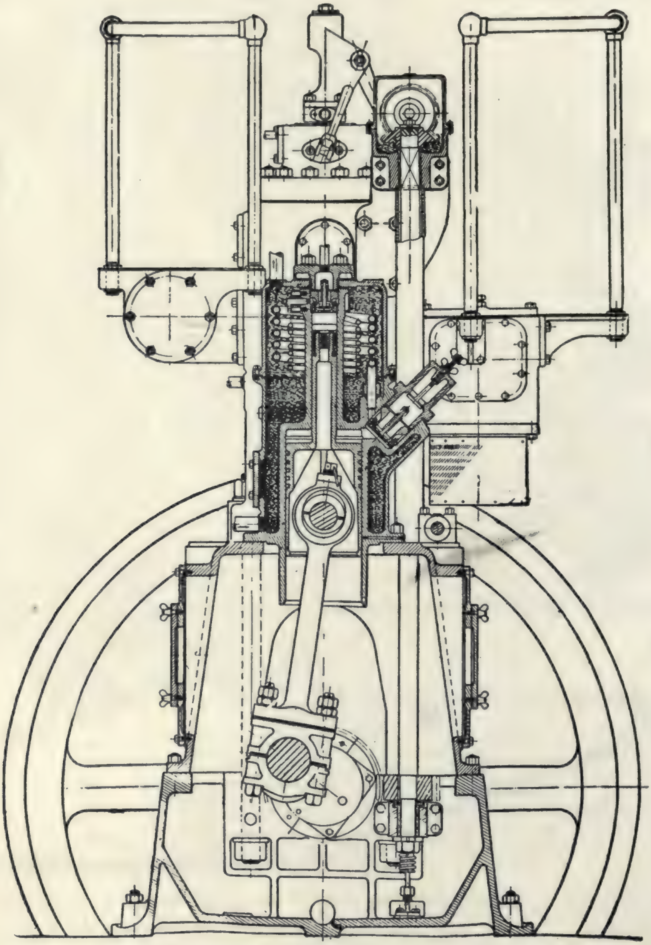
In addition to the large cover plates at front and rear, smaller inspection plates are also provided through which the operator can conveniently ascertain the temperature of the various bearings.



View With Portion of Railing Removed, Showing Cam Shaft. Indicator Rig, etc.

Air Compressor: The high pressure air required for injecting the fuel into the combustion space is furnished by a two stage compressor which is mounted on one end of the box frame and is driven directly from the crankshaft. A stepped piston is used to obtain the necessary displacements for the different stages. The connecting rod and its bearings are of very liberal proportions. They conform in design to the similar parts for the main working cylinder.

Both the low stage suction and discharge valve are located in a single pocket at the side of the cylinder. This pocket is so positioned as to make it impossible for a valve to get into the cylinder due to either a broken valve spring or stem, and yet the clearance volume is less than two per cent of the cylinder displacement.



Section Through Air Compressor Cylinder

The valve cage for the high stage valves is located in the upper end of the cylinder bore and with its cover forms the cylinder head. This cage is also arranged so that there is no possibility of a valve getting into the cylinder and wrecking the machine.

After being compressed in the first cylinder and before going to the second, also after the second compression the air is thoroughly cooled. The inter and after coolers consist of coils of copper tubing and are enclosed in the water jacket space of the compressor cylinder through which all the cooling water passes before going to the main cylinders. With this arrangement there are no hot pipes for the accumulation of carbon deposits and the high pressure air as it leaves the compressor is at practically the temperature of the inlet cooling water. Also after each compression provision is made for separating any entrained oil vapor or moisture from the air.

Scavenging Air Compressor: The air pressure for scavenging the main working cylinder is generated in the annular space above the crosshead and scavenging piston and around the main piston barrel. Because of its small clearance, the volumetric efficiency of this pump is well over 90 per cent, and as the area of this annular space is considerably greater than the area of the main cylinder, an ample quantity of pure air for cleaning and filling the working cylinder is insured.

The valves for both suctions and discharge are of the plate type and operate under very slight differences of pressure. As the areas through the valves and through all ports and passages are very liberal, the scavenging air is handled with the minimum amount of work, and as a result the overall mechanical efficiency of the engine is high, comparing very favorably with the best four-cycle engine practice.

Scavenging Manifold: The scavenging manifold which extends across the front of the engine contains the pockets in which are located the suction and discharge valves of the scavenging pumps, also the passage through which the air is distributed to the working cylinders. This passage is of ample area so as to cause the minimum of air friction. A large plate directly over each valve pocket permits of easy access to the valves. To the bottom of scavenging manifold is fixed the suction pipe through which the air is drawn to the scavenging pump. The sound of the air suction can be quite effectively muffled by means of a screen at the end of the pipe or if desired the pipe can be continued to the outside of the building.

Exhaust Manifold: The exhaust manifold is thoroughly water jacketed for its entire length, thus preventing heating up the engine room in warm weather. The passage for exhaust gases is of very liberal area so as to cause no harmful back pressure on the engine.

Fuel Pump: The fuel measuring pump is of the variable stroke type, a separate plunger is provided for each cylinder. The plunger stroke is controlled automatically by the governor. This governor is ordinarily set to give a total speed variation of 4 per cent between full and no

load, but is capable of considerable closer regulation if conditions necessitates the adjusting of the same.

Fuel Injection Valve: The fuel valve is of closed nozzle design. The proportions of the various parts have been determined by careful experiment so as to have the fuel not only thoroughly broken up into small particles, but also to have it well mixed with the injection air. That this result has been attained is evidenced by the low fuel consumption and clear exhaust at the various loads and with fuel and crude oils of widely varying characteristics. The needle valve is made from Tungsten steel carefully hardened and ground. It is thus well suited to resist both the chemical action of impurities in the oils, and mechanical wear of the packing.

Camshaft and Gears: The camshaft is mounted in brackets supported from the cylinder jackets. It is so located that the fuel valve can be operated directly through a single bell crank, but it is so positioned that it does not have to be disturbed in any way when removing a cylinder head or piston. It is driven from a vertical shaft by a set of mechanical miter gears which are cut from steel forgings. The vertical shaft gets its motion from the main shaft through a set of spiral gears.

Air Starting: For starting, compressed air is admitted to each working cylinder through an automatic check valve in the cylinder head. The air is properly timed by individual starting valves located just below the camshaft and each in front of the cylinder which it supplies. These valves are put in operation by the starting air pressure and automatically go out of operation as soon as the starting air is shut off at the main control valve. The engine can be started on an air pressure as low as one hundred and fifty pounds.

Lubrication: The lubrication of the Standard engine is accomplished by a force feed oil pump. The feeds to the various parts are from centrally located sight feeds, where the operator may at all times be enabled to observe proper functioning of the system.

EFFICIENCY PERFORMANCES OF STANDARD DIESELS

The first two charts in figure (a) show average Diesel engine practices for both the two and four-stroke-cycle types, and the third shows the distribution of work in the Standard engine when operated on 250 R.P.M. and rated 50 B.H.P. per cylinder. The low percentage of work for operating the scavenging pumps is, as already mentioned, largely due to the employment of an exceptionally low scavenging air pressure. While the reduction in work required for driving the injection air compressor is attributed to a design of atomizer in which not only is the pressure required very moderate, owing to a minimum number of restricted passages, but also the volume of injection air required is considerably reduced.

The slightly lower mechanical losses are explained by the ideal conditions, both as to temperature and lubrication, under which all bearings and rubbing surfaces operate.

In Figure (b) are reproductions of indicator diagrams taken at various loads with the engine operating at 250 R.P.M. The curves in Figure (i) show the results of fuel consumption tests on the 100 H.P. two-cylinder machine when operated at different loads and speeds. As would be expected, owing to the longer time allowed for combustion and the

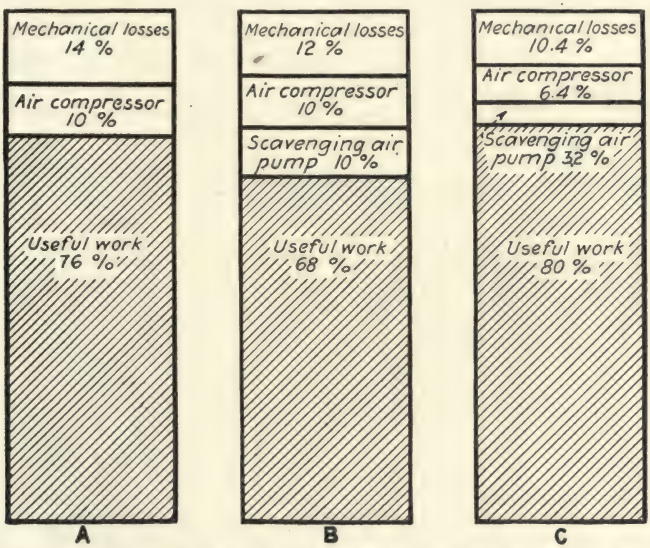


Figure (a) Comparison of Mechanical Efficiencies. A—Usual Four-Cycle Diesel; B—Two-stroke-cycle Diesel; C—Standard Two-stroke-cycle Diesel

slightly higher mechanical efficiency, somewhat better results are shown at the lower speeds. However, that the difference is as slight as shown, speaks very well indeed for the perfection of the design as to these two features.

When comparing these fuel consumption figures with others published, it should be borne in mind that they are for a cylinder of but 10¾-inches bore. For a cylinder of twice this diameter, which comes nearer the sizes on which figures are usually found, the results should be nearly 10 per cent lower.

In disapproval of the statement sometimes made that owing to the more frequent power impulses, the two-stroke-cycle engine operates with exceedingly high temperatures, the following figures as to exhaust temperature of the Standard engine are cited. With the engine in each case developing 100 H.P., the exhaust temperatures were as follows: At 225 R.P.M., 345 degrees Fahrenheit; at 250 R.P.M., 325 degrees Fahr-

enheit; at 275 R.P.M., 315 degrees Fahrenheit; at 300 R.P.M., 305 degrees Fahrenheit. While the temperatures are exceedingly low at all speeds, it is found that for the same delivered horsepower the lowest temperature result from the more frequent and light power impulses rather than the reverse. This, the builders of the Standard engine state, is in accordance with theory and should be the case for a properly designed machine. For not only does the theoretical efficiency of the Diesel cycle improve as the size of fuel charge is reduced, but there is also a larger

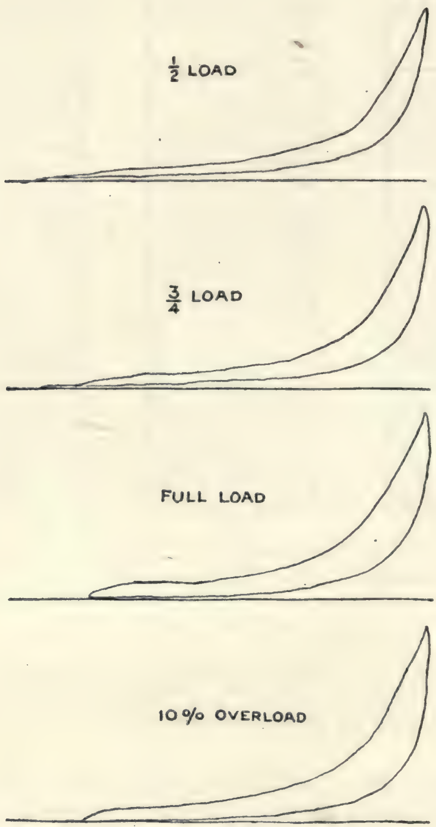


Figure (b) Indicator Diagrams at Various Loads

volume of cool air passing through the cylinder per unit of time. For the reasons cited they emphasize very strongly the desirability of employing lighter fuel charges and have adopted ratings requiring mean effective pressures of about 75 lbs., as against the usual Diesel practice of 80 to 100 pounds.

This, it is contended, is the logical manner for utilizing the admitted advantages of the two-cycle principle and of obtaining a machine which

with high thermal efficiency shall also combine the even more important requisite of unfailing reliability.

In the following description of the horizontal type of the product of the Hadfield-Penfield Steel Co., of Bucyrus, Ohio, a clear detailed information will be given of the Standard Horizontal Diesel Engine.

This particular type of engine follows the two-cycle principle, and is found satisfactory on horizontal construction. In summing up the advantages gained in two-cycle construction, following is claimed: For a given size cylinder the horsepower output would be nearly double and at the same time this result would be obtained with fewer working parts.

The frame is of center crank type with heavy box section walls. This casting also includes the scavenging air cylinder in which a differential or stepped piston operates, furnishing the air for cleaning and filling the main cylinder. Since the main or working cylinder extends into this larger cylinder for about one-half its length, the machine is practically all contained within this one casting, making a very massive and rigid construction.

The cylinder head is comparatively small and very simple casting of symmetrical design. There is but one opening, which extends through both walls; this is for the fuel spray valve and is located at the exact center. In addition to its symmetrical shape the casting is properly

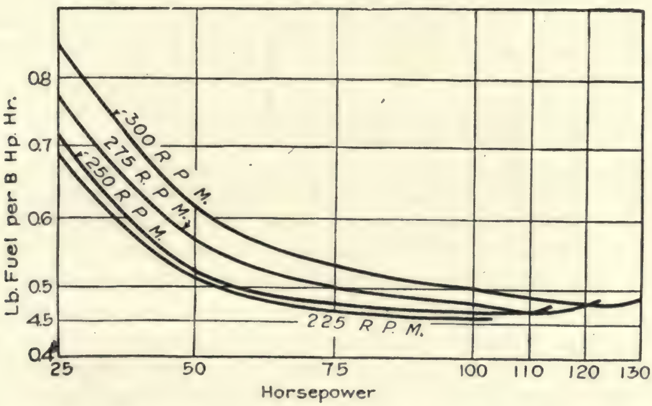


Fig. (c) Curves Showing Fuel Consumption at Different Loads and Speeds

water jacketed so that the danger of cracks developing is reduced to the minimum.

The combustion space which is formed by the inner wall is practically a half sphere, which is an ideal form, as it insures the most rapid mingling of the fuel spray with the air for burning, and hence early and complete combustion.

Close fitted piston and rings are features highly commendable. To make permissible these close fits, the pistons are water-cooled in all

except the very small sizes of cylinders. An overheated piston may develop cracks, may expand sufficiently to bind and score the cylinder, or may even stick fast, wrenching off connecting bolts or rod bolts and wrecking the whole machine. Water cooling eliminates the possibility of such troubles and at the same time makes the proper lubrication of piston much easier, and reduces the amount of oil required. As the piston pin is located in the larger air pumping piston, the main piston is relieved of all connecting rod thrust, and therefore the wear in the main cylinder is slight.

Identical two-stage compressor arrangement are on the horizontal as on the vertical types of Standard Diesels. The high pressure air required for injecting the fuel into the combustion space is furnished by a two-stage pump mounted on the side of the frame casting, and driven by a small crank on the end of the main shaft. The air from the first stage of this pump is delivered into tanks at about 150 pounds pressure, and is available for starting the engine. From these tanks the air is admitted to the second stage of the air pump and thence direct to the fuel valve without intermediate receiver. As the first stage of the pump takes its air from the scavenging air cylinder at about five pounds pressure rather than from the atmosphere, we have in reality a three-stage compression. This air pump is well proportioned for these types of engines and will furnish air, even if not properly kept up. The front end of the first stage piston has an extension in the form of a piston valve. This extension acts both as a crosshead for the air pump and also controls the suction and discharge of air from the large scavenging cylinder.

The fuel nozzle is of the open type; that is, the fuel is pumped into a small receptacle which is at all times in open communication with the combustion chamber. This fuel receptacle is placed just ahead of the mechanically operated timing valve which controls the admission of injection air. When the timing valve opens the fuel is picked up by the injection air and blown past the stationary atomizer into the combustion chamber. The open nozzle construction has the following advantages:

(1) The fuel pump has only to work against low pressures as it is timed to operate when the main piston is at the opposite end of the cylinder; the pump thus becomes in reality only a measuring device. The plunger can be packed very loosely and better regulation can be secured on account of the light load on the governor.

(2) As only pure air passes the timing valve, the seat will remain tight much longer than where air and fuel both are injected through the valve.

(3) Better atomizing is secured, as the heated air which is driven back in contact with the oil tends to vaporize it, and we do not therefore have to depend entirely upon mechanical action in breaking up the fuel. The valve cage, atomizer, etc., are made from nickel steel, as this material best resists the chemical action of the impurities frequently found in the crude oils.

These engines are guaranteed continuously without undue heating their full rated horsepower. The ratings given hold for an elevation of not to exceed 1,000 feet above sea level. For higher altitudes the capacities will somewhat decrease.

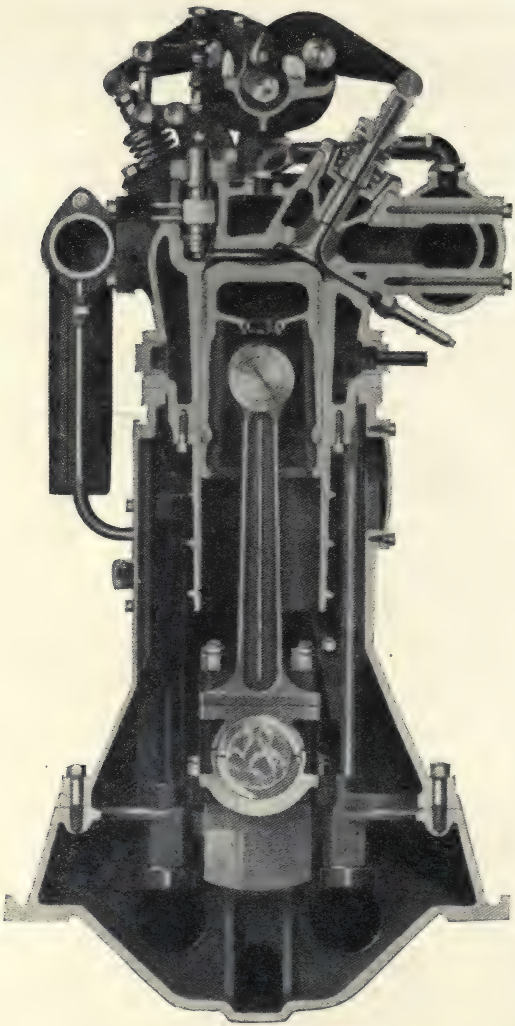
The Standard engine is offered subject to the guarantee that its fuel consumption shall not exceed following quantities of crude or cheap fuel oils.

At full load	.50 lbs. per B.H.P. hour.
At three-fourths load	.52 lbs. per B.H.P. hour.
At one-half load	.58 lbs. per B.H.P. hour.

This consumption is based on a sea level rating and a fuel of at least 18.000 B.T.U. (low heating value) per pound.

LOMBARD ENGINES

The cross-section views of the Lombard engines disclose the type of engine built with the object in view to act as a power producer where reliability is of highest importance. These engines are of the vertical, multi-cylinder, heavy duty type, with pistons, rings, connecting rods, bearings and crankshaft all easily accessible through the front housing doors of the enclosed crank case.



Cross-Sectional View of Lombard Engine.

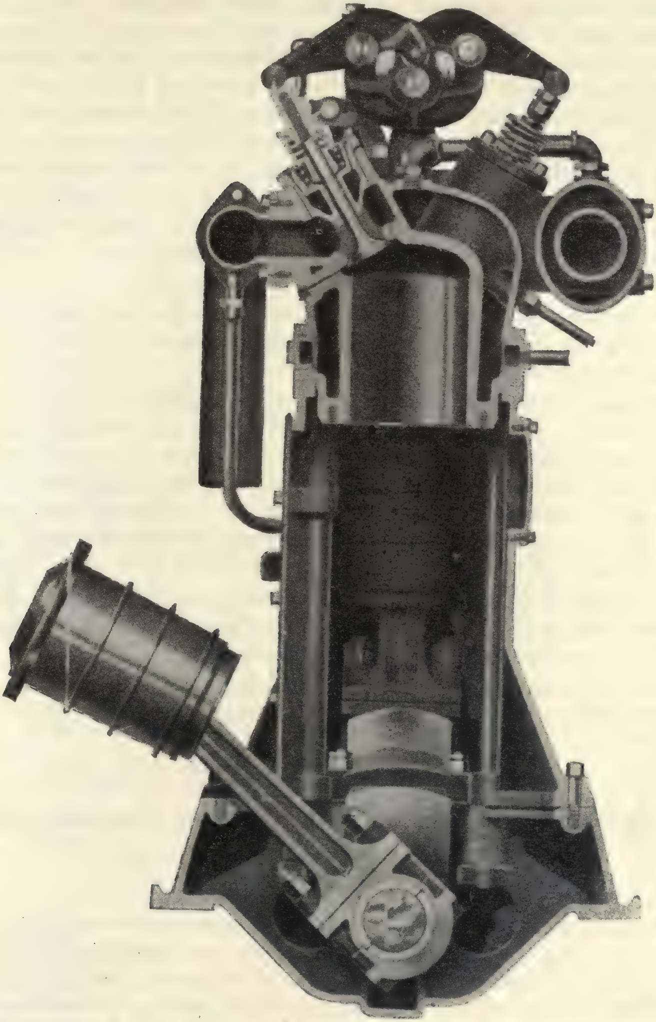


Illustration demonstrating the accessibility of Lombard's Vertical Multi-cylinder Diesel Engines.

This unusual accessibility results from the design of the cylinders with removable skirt section bolted to the bottom of each. With the crank on bottom center and skirt detached from cylinder casting, any piston with its connecting rod, can be swung forward through the crank case door opening, without removing cylinder head, disconnecting crank pin box, dismounting valve gear or otherwise disturbing adjustments which it is desirable to preserve.

Other advantages of this construction are that it (1) avoids the objectionable joint in the combustion chamber exposed to high tempera-

tures and pressures, the joint between cylinder and skirt sections being subjected only to exhaust temperatures and pressures; (2) eliminates the mass of metal necessary for a cylinder-and-head joint, thereby securing a much freer movement of the cylinder wall to take care of expansion and contraction and providing ample and unrestricted space for circulation of cooling water; (3) permits the accessible, overhead location of cam shaft with cams running in oil and with valves operated by simple rocker arms; and (4) results in an engine of pleasing, symmetrical appearance, with simplified controls for starting and handling a powerful, efficient unit, which saves weight, head room, installation and operating expense, upkeep, fuel and lubricating oil.

The engine, which is of multi-cylinder construction, are equipped with integral áir compressor of balanced duplex design for supplying the starting and injection air.

The frame sections which substantially enclose the crank case merely support weight. The four heavy steel tie rods, which extend from each cylinder to the main bearing pads within the bed plate, carry the working load and relieve the frame of all tensile stresses. Removal of frame housing doors and tie rods along the front or air intake side, permits rolling the main crank shaft out in the space directly alongside of the engine, without dismounting cylinders or disturbing valve gear.

All cylinders and bearing surfaces are positively and copiously lubricated from a pressure system which includes filters and coolers for the repeated and economical use of lubricating oil.

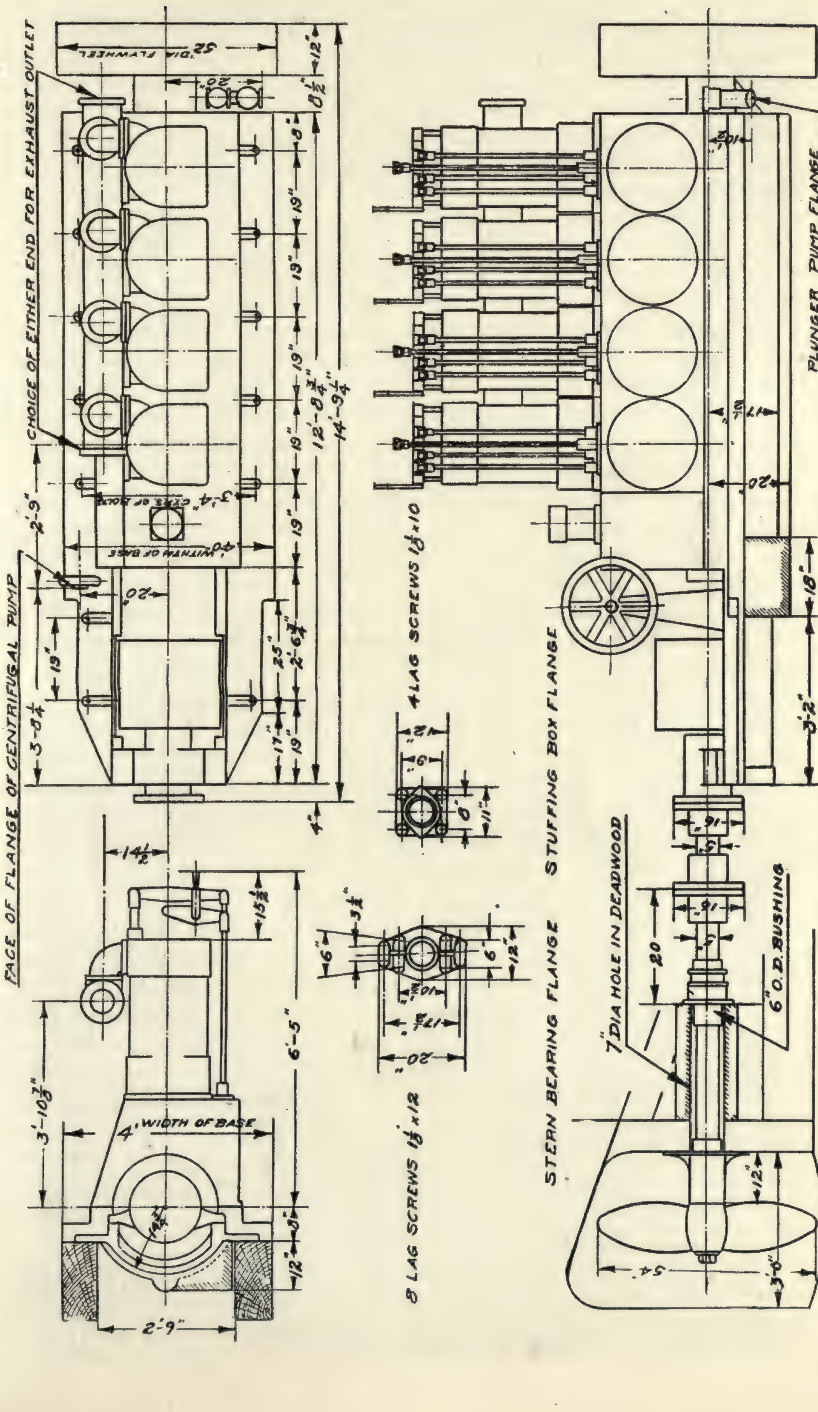
Lombard engines are built in sizes ranging from 60 to 500 B. H. P. Engines are built for either marine or stationary purpose.

ATLAS-IMPERIAL SOLID INJECTION DIESEL ENGINES FOUR-CYCLE MARINE TYPE

The rapid increase of small sized Diesel engines between 50 B. H. P. and 200 B. H. P. is primary due to the record accomplishments of this respective class of power producers. Specially adapted for coastwise service aboard ship as auxiliary driving mediums and on crafts mainly depending upon sails.

Their recognized ability as a dependable and economic machine, and above all the surprising simplicity in design and construction, not to mention the limited space a machine of this horsepower capacity occupies, makes this generator of Diesel power a favored type.

Firms, such as the Worthington Corporation, Western Machine Co., Dow Pump & Diesel Engine Co., Enterprise Diesel Engine Co., Lombard Governor Co., Atlas-Imperial Engine Co., and in fact the many manufacturers throughout the United States as well as Europe have made it possible to convince the shipowner that Diesel power is as profitable for small sized vessels as for ships of large carrying capacities.



Principal Dimensions for Installing Four-Cylinder, 165 H.P. Atlas-Imperial Mechanical Injection Diesel Marine Engine.

It will be acknowledged that only a few years ago numerous objectionable features were reacting to the detriment of adopting Diesel machinery in smaller crafts. The principal reason bringing the Diesel engine as an unfavorable machine for use on coastwise marine service or aboard ship as an auxiliary operating engine was the lack of skilled operators sufficiently acquainted with the principles and mechanism of Diesel power. That this very fact contributed greatly in retarding the adoption of this prime mover is true to a certain extent. Some of the earlier types of Diesel engines were rather crude and possessed of entirely too many contrivances, such as valves, piping, etc., which were often the cause of breakdowns with its consequential expensive loss of time to the owner. In many cases a special kind of fuel for the engine was necessary, which in itself proved disadvantageous. From a mechanical standpoint they were often inefficient and must be acknowledged that the existing difference of the modern types of Diesel engines in comparison to those of only a few years ago show the enormous strides which have taken place to bring the Diesel engine to its high stage of present day perfection.

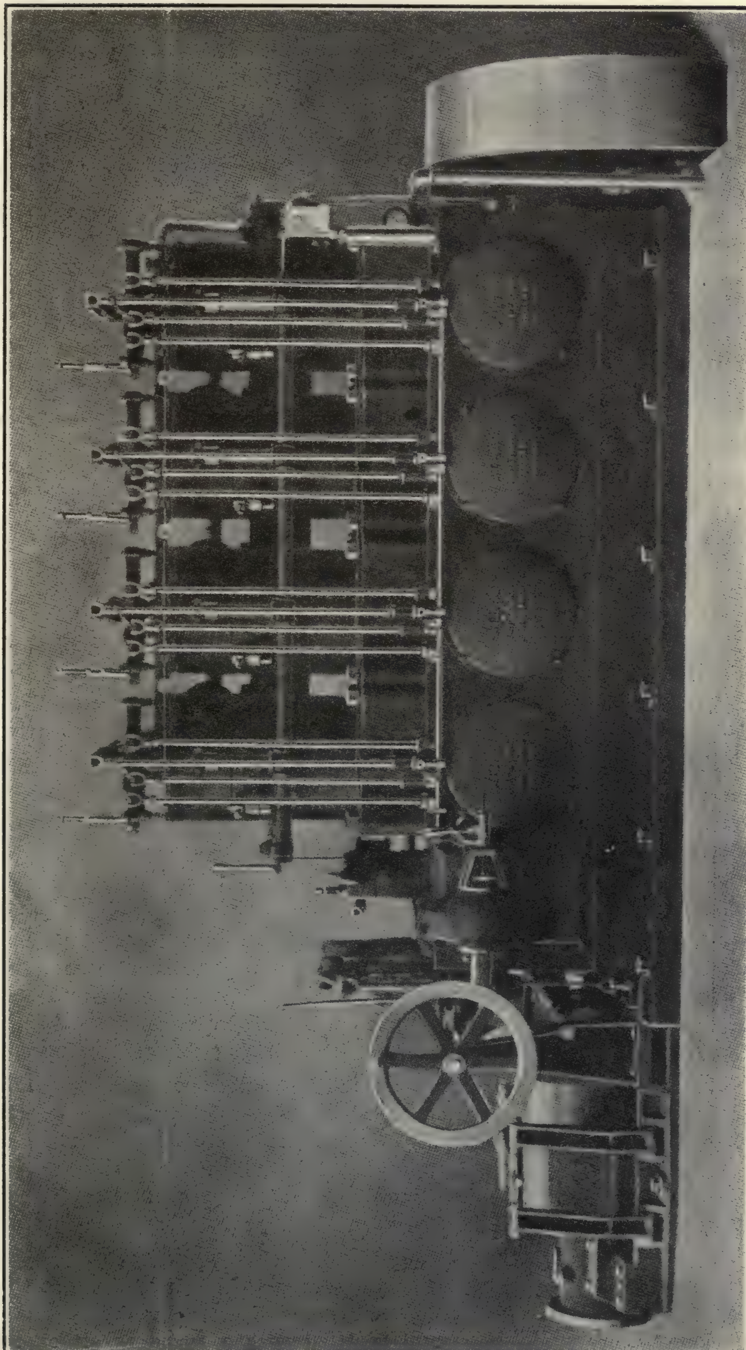
As will be observed in the specially selected types of Diesel machinery in this work, that the simplicity in mechanical arrangements have been brought to a stage that the person with limited mechanical knowledge around Diesel power can be entrusted with the operation of a machine, providing however, that his knowledge comprises the necessary ability upon which the fundamental laws of Internal Combustion machinery exists.

Often, troubles which arise around Diesel machinery have been unjustly blamed upon the machine and its manufacturer, designer, etc., when as a matter of fact the engine has stood an unmerciful test before being installed. After some investigation it usually results in the establishment of the fact that the operator was in an entire wrong place and in justice to the many experienced operators ought to retire from his chosen profession or occupation.

Much could be said on this subject dealing with the incompetency of the man in charge, but, it will be admitted, that with the introduction of literature on the subject of Diesel operation and the greater opportunity afforded to day to receive a better training in this branch of engineering, will contribute towards the creation of better class of men.

When carefully giving the Atlas-Imperial Diesel engine some study we find that there are points which deserve of highest comment. The writers of "The 20th Century Guide for Diesel Operators" show an entirely impartial conduct, and it is to be hoped that the reader will not labor under an illusion that any special favors are granted the numerous firms which are giving space in this work. As previously stated, every firm illustrated in this book are selected by their merits and are known the world over as reliable manufacturers.

The Atlas-Imperial Diesel engine is of the four-cylinder, four-cycle vertical type, having enclosed crankcase, valves in the head, fitted with heavy duty reversing gear, and force feed lubricating-system.



Full View of 165 B.H.P. Atlas-Imperial Four-Cycle Mechanical Injection Diesel Engine.

To the operator experienced with gasoline-driven engines, the "valve-in-head" arrangement will sound familiar. It will be of interest here to give a little explanation of the advantages claimed by builders of valve-in-head motors in contrast to L-head or T-head types of construction.

Unfortunately it is impossible for a Diesel engine to utilize all the heat created, or rather generated for power. If some means were not adopted to cool the motor the heat would become so great that it would be destructive to the motor.

So in making the cylinder castings, water passages are cast around the cylinders in such a manner as to allow the excess heat to escape through the cylinder walls, into the water, which in turn is cooled by the circulation method of the engine. It is quite evident, therefore, that the less water jacketing space there is in a motor, the greater the thermal (heat) efficiency there will be because of smaller area of the cylinder walls and combustion chamber will be exposed to the cooling influence of the water.

This brings us to the biggest reason for the Valve-in-Head design, because the arrangement of the valves permits of a smaller, more compact combustion chamber than is possible in either the L-Head or the T-Head type of engine. To make this statement still clearer, it should be understood that in all cases, both inlet and exhaust valves form a part of the combustion chamber, where the heat is the greatest, and in consequence it is necessary to provide ample water jacketing space to the heads and sides of the cylinders. In this engine this is accomplished by means of passover pipes causing the circulation of water.

Now, if we regard our fuel-oil as so many heat units, it is quite apparent that the loss of these heat units that are wasted through the water jacketed surfaces, the more of them will be left in the form of actual, usable power directed against the piston.

Then, because of the larger valves this type of construction permits and can be located in a straight line above the pistons, the dead exhaust gases are quickly and easily expelled through them at the conclusion of the working stroke, instead of being forced around corners and downward through a much larger chamber, as in the L-Head and T-Head types. And the combustion during each working stroke is much more perfect in this type of design because the incoming charges are purer.

The exhaust and inlet valves are mechanically operated. The valve head is made of cast iron with steel stem. The inlet and exhaust valve springs are interchangeable. The valves are operated by steel valve lifters provided with large anti-friction rollers, these rollers are made of special alloy steel, hardened and ground. The cams are fastened to the camshaft with keys. The cam shaft gear is 16" diameter and 2½" face.

The engine is fitted with a flyball governor which is of the throttling type. The governor is driven by spur gears direct from the cam gearing without belts or frictions and is fitted with a speeding attachment

whereby the speed of the engine may be changed as desired at any time while the engine is in motion. The governor acts directly on the spray valve lifters and controls the amount of fuel delivered to the spray valves in direct proportion to the power developed. These governors are very sensitive and quick of action.

The engine is provided with a force feed lubricating system which delivers oil at about 5 lbs. pressure to all the main working parts. The oil is delivered from the pressure pump to the main crankshaft bearings, from which it passes through the hollow crankshaft to the crankpin bearing, and from there it passes up the hollow connecting rods to the piston pin bearing. The oil is then returned by means of a sump pump to a strainer from which it again passes through the pressure pump. In addition to this system the engine is provided with a multiple force feed lubricator connected with copper tubes to the cylinders, etc.

The mechanical injection fuel pumps, which are made of steel, are fitted to the engine. These pumps deliver the fuel under pressure to the spray valves in the center of each cylinder head.

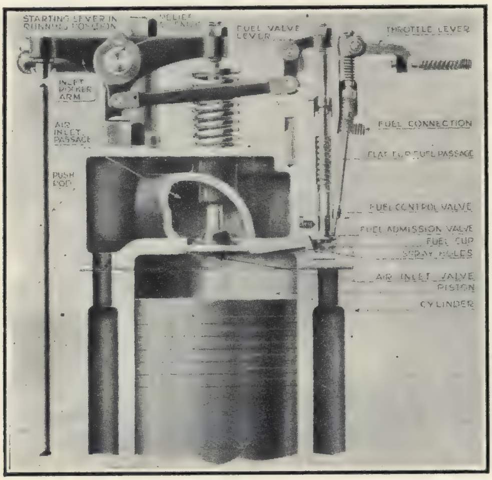
THE CUMMINS OIL ENGINE

(Diesel Type)

Note: The Cummins Oil Engine is operated under the four-stroke-cycle system. While the engine resembles in its method of power generation the Diesel principle, nevertheless, when carefully studying this machine, it will be found that it has exclusive features, which makes this engine a distinctive type of its own.

Principle of Operation: (1) The principle of operation is briefly as follows:

On the first stroke as the piston descends a charge of pure air is drawn through intake valve directly from the outside; at the same time the same mechanism pushes open small fuel valve in fuel injector body and permits a charge of oil to pass into cup. This amount of charge



Sectional View Through Cylinder Head of Cummins Four-Cycle, Valve-in-Head, 8 to 32 H.P. Diesel Engines.

which determines the speed, is controlled by the hand throttle operating on needle valve.

(2) At the end of this first stroke the intake valve closes and the piston comes up, compressing the charge of pure air to the pressure of approximately 450 pounds per square inch.

(3) This pressure causes the air to instantly become heated to a temperature of approximately 1000 degrees F., or in other words, practically red hot.

(4) As the temperature and pressure in fuel cup is the same as that in the cylinder, the fuel becomes highly heated and the small amount of gas which is in the fuel charge ignites, raising the pressure in the cup to a point considerable higher than that outside in the cylinder.

(5) This difference in pressure causes the heavy oil left in cup to be blown out into the cylinder in a fine gaseous spray which ignites instantly and causes the expansion or working stroke of piston.

(6) On the fourth and last cycle, the exhaust valve opens and the burned gases are expelled through exhaust valve into the atmosphere.

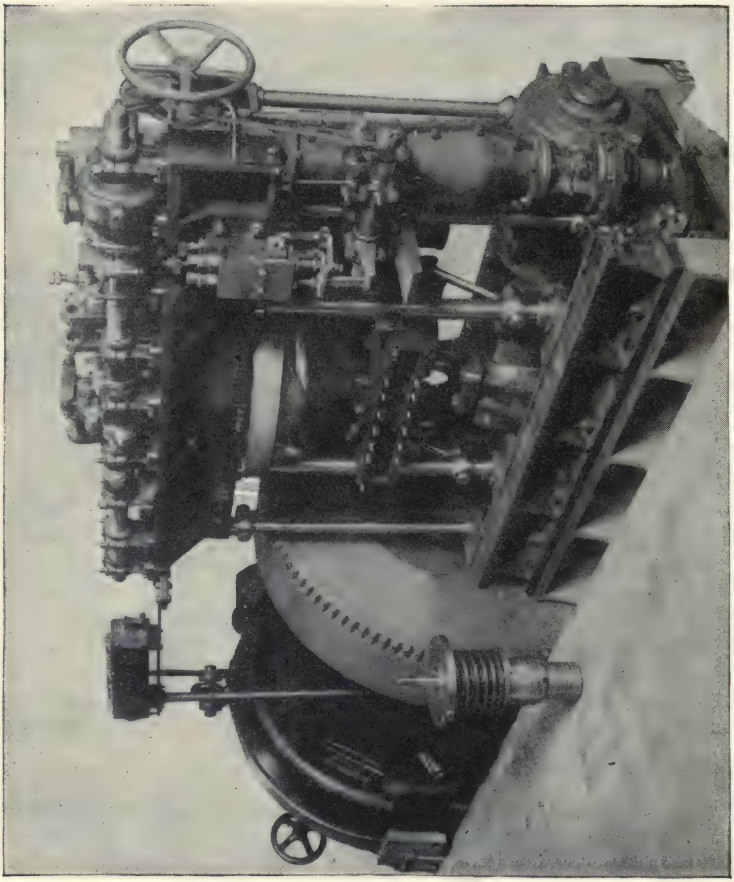
A study of the sectional view of the illustration of the Cummins Oil engine should clarify this mode of generation.

(7) The fuel enters the fuel valve body at a point marked "Fuel Connection," passes down around needle valve, which has three flat sides, permitting free flow of fuel. In the lower end of this needle valve a long hardened and ground taper seat fits into the seat in fuel valve body. At the top of fuel valve a brass cage carries fuel valve lever and throttle lever. The throttle lever works a double thread screw which permits spring to lift needle valve off seat according to opening of throttle lever.

(8) At the beginning of the suction stroke the air-inlet valve is opened by the push rod, which causes the small lever pinned to the rocker arm to in turn open the fuel-admission valve; this permits the proper amount of fuel, which is determined by fuel-control valve, to be drawn into fuel cup, where it lies until the heat of compression on compression stroke ignites it.

A study of the cut will show that the fuel cup is only exposed to the heat of combustion chamber for a very small margin around spray holes. This is a vital feature exclusive of this type of engine.

Starting: An eccentrically operated compression release is fitted to valve rocker shaft. By throwing up small lever, eccentric inside rocker arm is lowered, holding exhaust valve open. This permits engine to be revolved by starting crank on end of crank shaft with no resistance except small amount of friction in engine. After spinning engine over rapidly several times, the lever is dropped, which causes exhaust valve to seat properly, and the heavy flywheel carries engine over its first compression. This fires the fuel charge which was admitted through the valve and the engine starts firing.

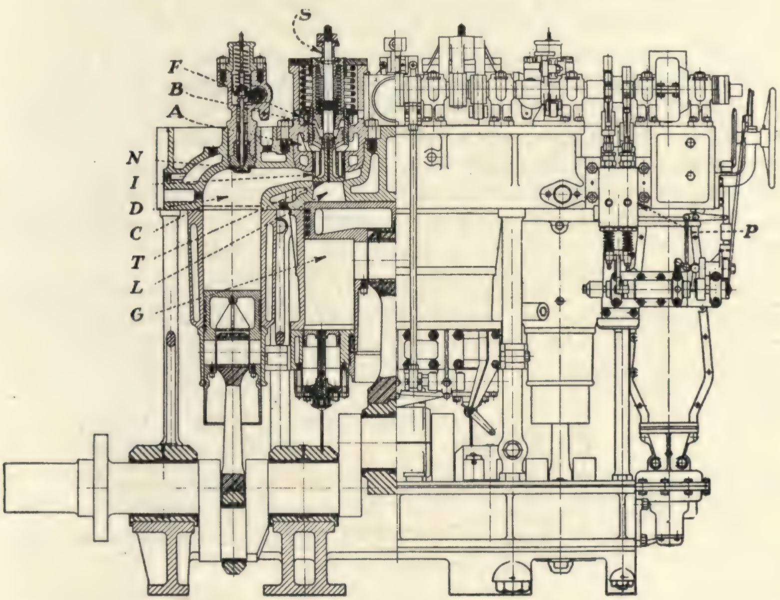


Full View of Sperry Compound Engine. Note Two "High Pressure" (Four-Stroke-Cycle) and One "Low Pressure" (Two-Stroke-Cycle Cylinder

SPERRY'S MARINE TYPE HEAVY DUTY COMPOUND DIESEL ENGINE

The principle on which this engine operates differs vastly from any motor now on the market. It is very ingenious in design, and experiences with the first engines in coastwise service has demonstrated its suitability and economy equal to the best of Diesels of same horsepower capacity.

The illustration shows a small marine type with high-pressure cylinders 7 inches by 11 inches, running at 400 R.P.M. The fuel pumps are also shown here and the connection to the governor. The camshaft is



Sperry Compound Engine, Cross-Sectional View. Letters Indicating Constructive Features of Transfer-Valve, Port Arrangement, Pistons, etc.

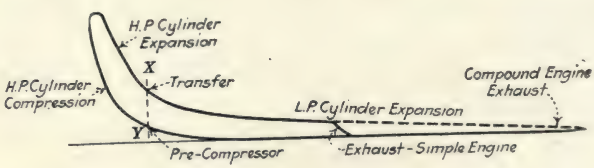
on a shelf at the top of the engine to one side and is driven by skew-gears. The electric generator forming the full load of this engine is shown in the background and one of the transfer valves with its bonnet cover stands on the floor in front of the engine. The comparatively small size of the engine should meet with the approval aboard such vessels where small space allowance is of essential importance. The construction is shown very complete in this illustration, especially the accessibility and similarity to ordinary types of engines in regards to construction.

Construction and Mechanical Efficiency: It remains to be seen, if the highly commendable efforts of Mr. Elmer A. Sperry, M. E., has by the creation of this latest addition of improved Diesel type solved the question of eliminating excessive weighty Diesels and substituting in its place the much lighter "compound" type. If this has been accomplished an added interest will be paid to the future development of this new type in larger construction.

Its factors of established high mechanical efficiency in compounding Diesels we will undertake to briefly explain.

We are brought face to face with steam engine practice as prevailing in compound engines. Compound engines are a type where the high-pressure is taking up before entering the low pressure cylinder. Inasmuch as the Diesel engine is a "constant pressure" engine, where the larger volume of power gases in the combustion chamber of the compound at once solves a number of important problems, makes the light engine easy of accomplishment, and overcomes a number of difficulties at the same time. In this engine we have two high pressure or combustion pistons at the ends and a low pressure in the center. A balancing cylinder sustains a permanent connection with the low pressure cylinder. The solid fuel injection valve and nozzle are placed approximately over the center of gravity of the large masses of air in the so-called clearance dome.

It is understood that the two high-pressure cylinders are operating four-cycle, one 360 degree back of the other, discharging alternately into the low pressure, which therefore works two-cycle and delivers power on each down stroke.



Comparison in Compound Indicator Card in contrast to Diesel standard type.

The dome, unlike the usual type of Diesels is rather large and forms an upward extension of the combustion cylinder, extending also to the right in a large sweep surrounding a so-called "transfer" valve which seals the transfer port. A sleeve-like induction valve, seated on top of the transfer valve, is controlled by a cam-operated fork. The transfer valve and sleeve are lifted by a fork, located in a thimble near the top of the stem. The first-stage annular compression pump surrounding the trunk piston below the low-pressure piston proper, delivers its air to a small receiver, which in turn discharges to the cored port surrounding the induction sleeve.

The cooling is effected by following method: In forcing the high-pressure piston down air must pass some port in entering. The air-cooling port is in line with the transfer port and the induction valve itself rides on the back of the transfer valve in the form of a hollow sleeve seated directly on the top of the transfer valve. The back of the transfer valve is provided with greatly enlarged radiating and cooling surfaces presented to this cooling air and powerful convection currents are constantly acting when sealed. Moreover, this air when entering is at high velocity and gushes down through and bathes the deeply serrated surfaces of the back of the transfer valve, licking up the heat very completely in its inward rush.

THE STILL ENGINE

(Author's Note.—This article is a contribution through courtesy of the distinguished Professor of Naval Architecture and Marine Engineering at the Lehigh University, Bethlehem, Pa., Mr. L. B. Chapman.)

As will be observed by the detailed description of the Still engine that it is a combination Diesel and steam engine, devised to increase the thermal efficiency over that of the Diesel engine. The heat ordinarily rejected in the jacket water and to the exhaust is used to produce steam and about 8 per cent of this heat is converted into useful work, increasing the brake horsepower of the engine about 30 per cent.

Whatever the future of this engine may be, it must be considered from the standpoint of technical observation a factor of highest consideration. When giving this engine some study, it will be found that there are advantages, in particular in lessening fuel expenditure, which are deserving of comment.

From the diagram of the Still engine shown in Fig. 1 it will be seen that in its present form the engine is double-acting with the Diesel cycle working on top of the piston. The water jacket is connected in a circuit with the boiler and an exhaust generator as shown in the diagram. The cooling water enters and leaves the jacket at a constant temperature corresponding to the pressure of the steam in the boiler. The heat absorbed by the jacket water surrounding the combustion cylinder is used to convert the water into steam at constant temperature. In other words, the heat of combustion that radiates through the walls is transferred into latent heat of steam. The steam thus generated passes to the boiler. The function of the boiler is to produce steam for warming up and starting the engine and to augment the supply generated in the jackets, if the jacket supply is not sufficient for the steam end of the cylinder.

The boiler feed water is circulated through the jacket as shown in Fig. 1. The feed water is taken from the feed tank by the feed pump as in all steam plants and is delivered at about 100 degrees to a feed heater or exhaust generator where it absorbs the heat in the exhaust

gases. The temperature of the feed water is thus raised from 100 degrees Fahr. to between 350 and 450 degrees Fahr., and the exhaust gases are reduced from 900 to 150 degrees Fahr. The feed water then enters the jacket, where it is converted into steam by the heat of combustion.

The steam from the boiler enters the lower part of the cylinder and acts on the piston in practically the same manner as in a steam engine and is then exhausted to the condenser. Cylinder condensation, which is a large loss with the ordinary steam engine, is practically eliminated in the Still engine because of the heat received from the combustion of the gases in the Diesel end.

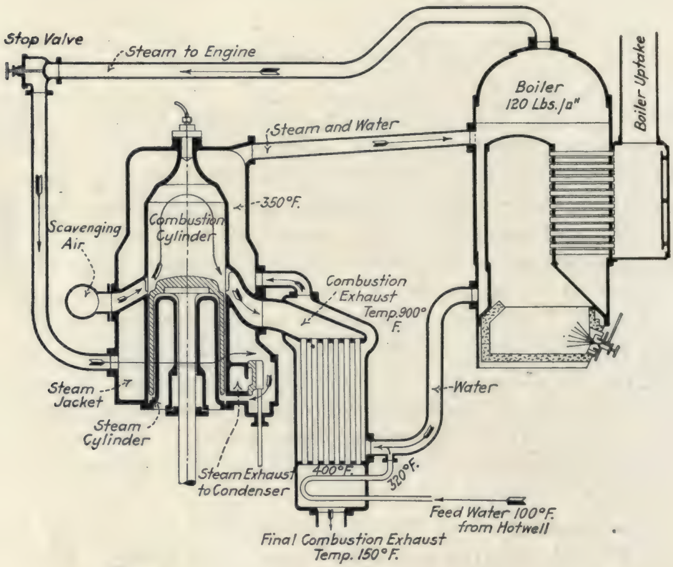


Fig. 1

During compression of the air on the Diesel side of the piston the air charge absorbs heat from the cylinder walls because of the high temperature in the jacket. With the straight Diesel engine the transfer is in the opposite direction, due to the cold circulating water. One result of this is that the required compression pressure is less in the Still engine than in ordinary Diesel engines.

Advantages of Combined Cycles: The advantages due to the interaction of the combustion and steam cycles are summarized by Mr. F. E. D. Acland in a paper before the Royal Society of Arts as follows:

(1) The mean temperature of the cylinder walls is higher than in ordinary engines; the cooler parts being maintained at a higher, the hotter parts at a lower temperature.

(2) The piston is cooler, owing to the expansion of the steam behind it.

(3) The heat efficiency of the combustion cycle is augmented owing to the walls being at a higher and constant temperature, and is in proportion to the rise in temperature of the jacket water.

(4) Frictional losses are reduced by the higher temperature, and by the steam overcoming the inertia of the reciprocating masses at the lower dead center.

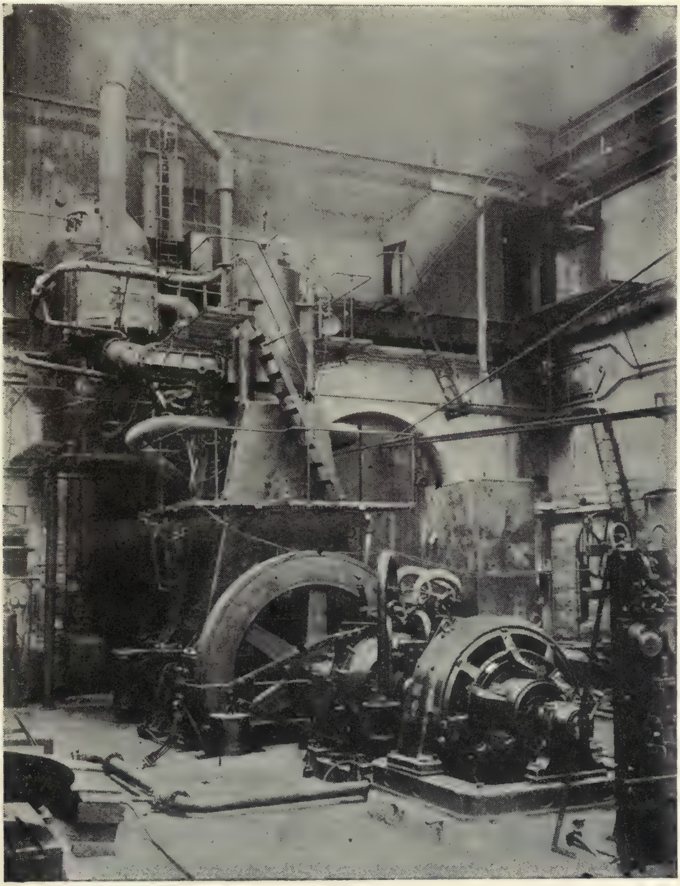


Fig. 2—Still Engine on Test in Shop

(5) The mechanical efficiency of the whole engine is higher than that obtainable in a normal engine of similar type.

(6) The steam, expanding as it does in a cylinder hotter than itself, gives an indicator diagram larger than that theoretically obtainable under ideal conditions in an ordinary steam engine.

(7) Twenty-nine per cent of additional brake horsepower is added to the shaft of the engine without increase in the fuel consumption. (Steam not condensed.)

(8) Forty per cent is added when condenser is used. (Air pump separately driven.)

(9) The indicated horsepower due to steam appears as brake horsepower added to the shaft, all the mechanical losses having already been accounted for in measuring the combustion brake horsepower.

Besides the merits listed above the two-cycle Still engine has the following advantages:

(1) Fuel consumption 10 to 20 per cent lower than the Diesel engine.

(2) Absence of cold circulating water causing large temperature difference and trouble with cylinder and head castings.

(4) Lower compression pressure.

(5) Absence of air-starting, circulating and piston-cooling system.

(6) Absence of exhaust valves and gear.

(7) Increased horsepower for a given bore and stroke.

(8) Possibility of overload by forcing steam boiler.

(9) Maneuvering at low revolution per minute is possible.

(10) High temperature range 2000 to 150 degrees F. (Carnot efficiency).

Trials with a single cylinder two-cycle Still engine were carried out during 1921 by Scott's Shipbuilding & Engineering Company, Greenock, Scotland. Owing to the fact that the experimental engine had only one cylinder, it was necessary to provide a small auxiliary high pressure steam cylinder, the lower end of the main Still cylinder serving as a low pressure steam cylinder. In an actual installation where several cylinders are used the lower part of one can be used as a high pressure cylinder, and the lower part of another as a low pressure cylinder, thus obviating the use of an auxiliary cylinder. All the auxiliaries, except the scavenging air pump, were driven off the main engine in these trials.

A photograph of this engine is shown in Fig. 2, and a digrammatic view showing all the auxiliaries in Fig. 3.

The result of the trials of this engine are given in the following table:

TRIALS OF 22-INCH BY 36-INCH STILL OIL ENGINE:

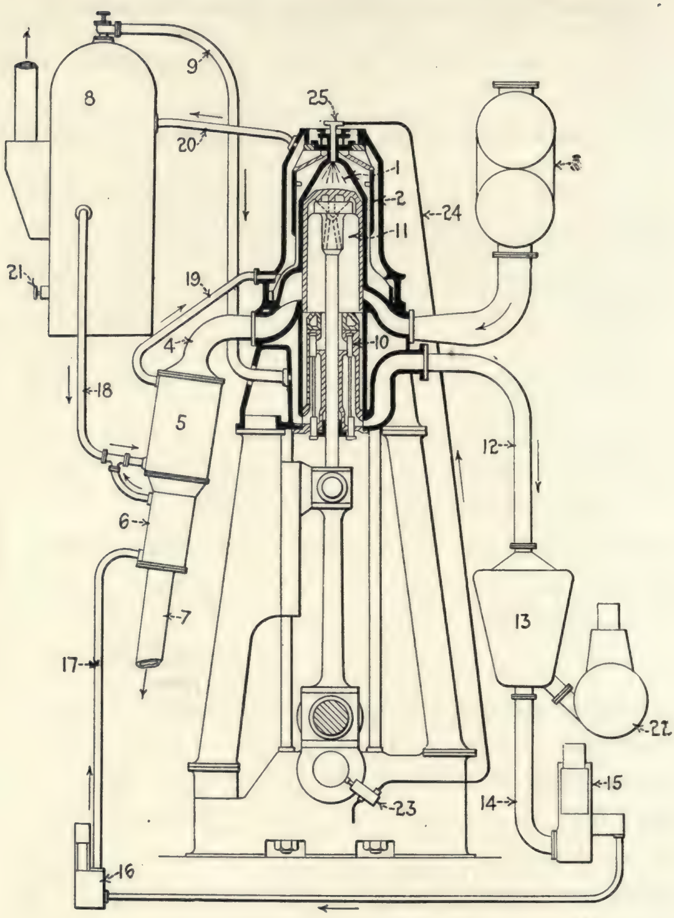
Main Still cylinder—Stroke, 36 inches. Bore, 22 inches. Piston rod, 6¼ inches.

Auxiliary high-pressure cylinder—Stroke, 14 inches. Bore, 22 inches.

		Over-load	Full load	Half load
1.	Average combustion M.E.P., lbs. per sq. in.	88.9	81.2	54.2
2.	Average steam M.E.P. referred to H.P.	4.43	3.80	1.26
3.	Average steam M.E.P. referred to L.P.	7.36	6.23	3.60
4.	Total M. E. P.	100.69	91.25	59.06
5.	R. P. M.	128.1	124.3	103
6.	Steam boiler pressure, lbs. per sq. in. gauge	112	100	108
7.	H.P. receiver pressure, lbs. per sq. in. gauge	75	57	23.5
8.	L.P. receiver pressure, lbs. per sq. in. gauge	11	5.5	0.4
9.	Vacuum, inches Hg.	28	27.5	26.6
10.	Water evaporated per hour, lbs.	950	807	388
11.	Scavenging pressure, inches water	49	46	40
12.	H.P. for scavenging	15.4	14.1	12.0
13.	Combustion I.H.P.	394	349 5	192.5
14.	Total I.H.P.	446	392	210
15.	Engine B.H.P.	384	343	174.5
16.	Net B.H.P. (line 15—line 12)	368.6	329	162.5
17.	Oil per hour, lbs.	146.6	123.4	64.0
18.	Oil per net B.H.P. per hour	.398	.375	.394
19.	Efficiency on net B.H.P., per cent	35.5	37.7	35.8

It will be observed that the fuel consumption at full load is 0.375 pounds per brake horsepower, which is about 10 per cent lower than the best four-cycle Diesel practice and nearly 20 per cent better than the general run of two-cycle Diesel engines.

Claims are put forward that the Still engine weighs less and occupies less space than the Diesel engine. The gain in economy of between 10 and 15 per cent for this single cylinder engine is highly encouraging and no doubt this can be improved upon when several cylinders are used. At first thought the engine appears complicated, but it must be borne in mind that the air-starting, circulating and piston-cooling systems are eliminated and the small boiler employed with the Still engine would generally be required on a Diesel ship.



Cross-Section of Still Engine

DEFINITION OF PARTS:

1. Combustion Cylinder.
2. Reinforcing Steel Hoop.
3. Scavenging Blower.
4. Combustion Exhaust Pipe Jacketed by Boiler Water.
5. Exhaust Generator.
6. Feed Water Heater.
7. Final Combustion Exhaust to Atmosphere.
8. Boiler.
9. Main Steam Pipe.
10. Steam Inlet and Exhaust Valves.
11. Steam Cylinder.

12. Steam Exhaust to Condenser.
13. Condenser.
14. Suction Pipe to Air Pump.
15. Air Pump.
16. Feed Pump.
17. Delivery Pipe to Feed Heater.
18. Circulating Water, Boiler to Exhaust Generator.
19. Circulating Water, Exhaust Generator to Cylinder Jacket.
20. Circulating Water and Steam; Jacket to Boiler.
21. Auxiliary Oil Burner.
22. Condenser Circulating Pump.
23. Oil Fuel Injection Pump.
24. Oil Fuel Delivery to Injection Valve.
25. Injection Valve.

THE WASHINGTON-ESTEP MARINE DIESEL ENGINE.

Four-Cycle Construction. Airless Injection. (Solid Injection).

The Washington-Estep Diesel Engine is a modern type of Diesel, following the design of medium sized power-producers particularly adapted for smaller crafts or for use as auxiliary machinery aboard ship. While this engine being built for marine purpose, the numerous novel features which this design embodies makes it an elegant factor for stationary work. We will define the special features of this latest type of Diesel engine.

The engine illustrated in figure (a) is of three-cylinder, four-cycle construction. Its horsepower rating ranges from 65 to 140 B. H. P., according to the specified size desired. The engine has a normal load of 100 B. H. P. at 280 R. P. M., when running with a continued M. E. P. of 85 lbs. per square inch. From tests at hand it performs power at 125 B. H. P. with ease and up to 140 B. H. P. as a maximum.

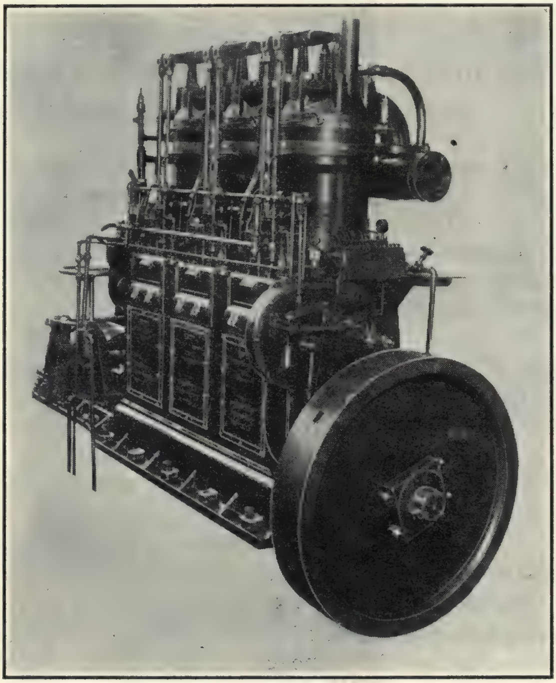
There are two fuel-injection valves of special design in each cylinder head, and an individual fuel pump supplies each pair of injectors with fuel for every cylinder. A governor controls the fuel at all speeds.

The designers have found it advisable to depart from the usual design of cylinders, having the same equipped with removable liners or bushings. Ample water jacket space is provided causing the thermal efficiency of the engine to be brought to a high standard. It will be realized, in comparing this engine with similar types of equal horsepower capacities, that the revolution performances is rather high, being from 210 to 350 revolutions per minute.

For marine service, where the average speed is generally up to 120 R. P. M., rarely exceeding 200 R. P. M., an engine of this nature must be capable to withstand the high thermal increase. All provisioncy to guard against this excess heat temperature have been made by the designer of this engine to insure ample lubrication. A double system, i.e., forced-

feed circulating system through manifold and crankshaft to all bearings and crosshead pins have been found imperative. A sump pump takes the return oil from the crank pits through strainer to auxiliary filtering and cooling tank, automatic governed pressure pump has been provided from this tank to bearings. A 10-feed mechanical oiler of the Manzel type furnishes fresh oil for cylinder lubrication.

The engine is fitted with bronze centrifugal circulating pumps and stand-by bronze plunger pump running half speed from cam shaft, which can be used for bilge and deck service. A water cooled air pump to supply starting air at 150 lbs. pressure is also provided. The fuel oil service pump is mounted on forward end of the frame.



Washington-Estep Engine, Port Side View

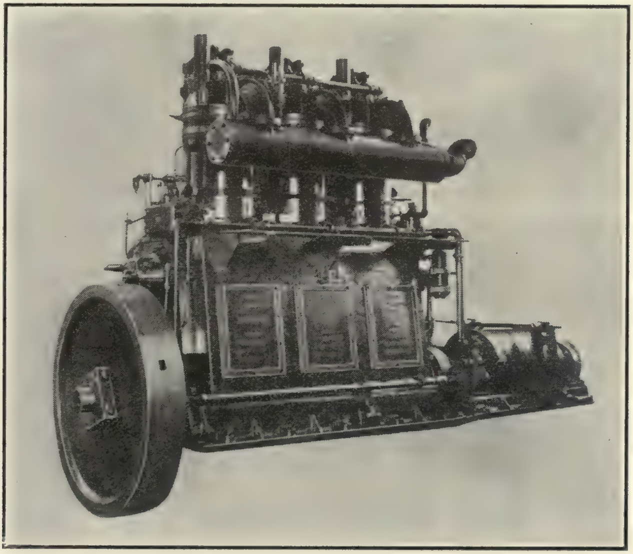
As costumary on small and medium sized engines, the piston is of the trunk type, convex head, strongly ribbed to support large hardened steel piston pin. The piston is equipped with six piston rings of special type for this service.

A commendable feature is the large cam shaft which is mounted in adjustable removable bearings, fitted with hardened steel accurately machined cams, all enclosed running in oil, driven with simple spur gears

of large diameter and ample face. The idler gear is bronze and running in oil. Compression release for hand turning is also provided.

The reverse gear is of a patented design by the builders of this engine. It is of heavy duty type, positive, simple, accessible and can be backed for hours without heating. The clutch is the latest multiple disc type enclosed, all gears are heavy simple spur type cut from high carbon forgings. The entire assembly can be quickly removed from engine by letting go the shaft coupling at each end.

The starting can be accomplished from cold, a simple water-cooled air compressor being mounted at the after end of the engine for the purpose of charging starting-air tanks at 150 lbs. pressure. The compression of the engine is limited to 350 lbs., being of similar figure to most solid injection types of engines of this capacity.

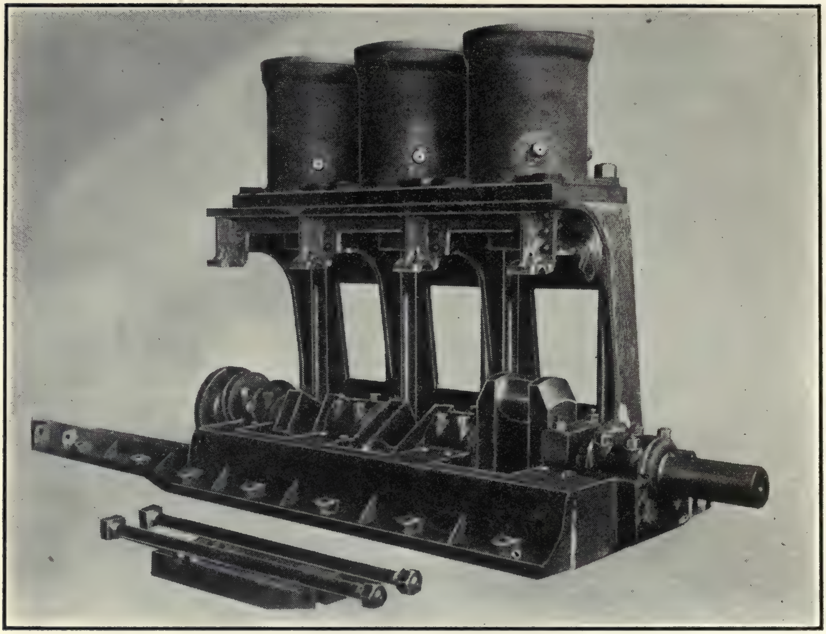


Washington-Estep Engine, Starboard Side View

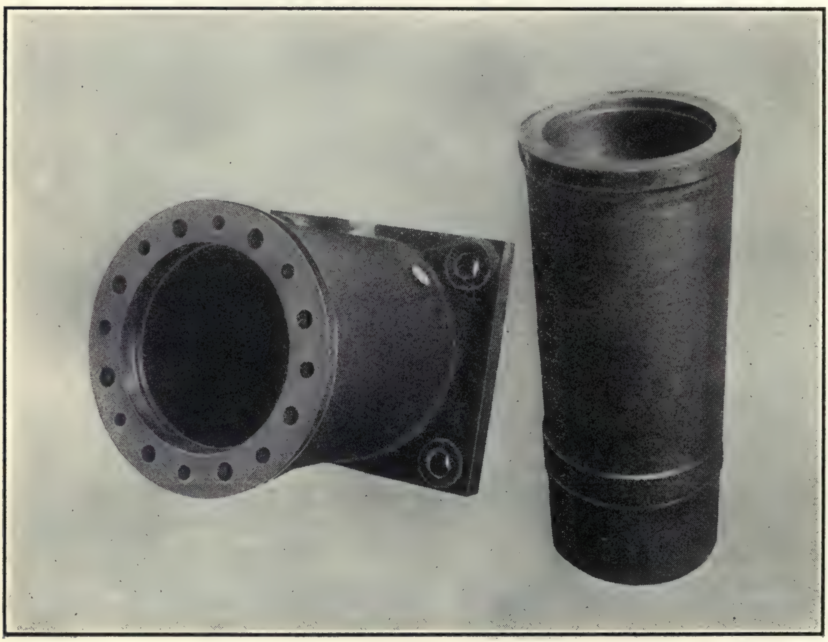
It will be mentioned here that instead of the high pump pressure usually adopted with airless injection, or direct or sometimes called solid injection, the designer has found it feasible to use a comparatively low pressure of 1,500 lbs.

The thrust bearing is of the removable type and is totally enclosed, water-cooled, and provided with a mechanical oiling system. All bearings are removable.

The high-economy establishment of this engine, brings it within reach of very limited operating expenditure. The engine will consume



Washington-Estep Engine, Showing Engine-frame and Cylinder Construction



Estep Design of Cylinders and Liners

5½ gallons of average calorific value of fuels at a cost of 63 cents per gallon, in addition to 1/10 gallon of lubricating oil at 40 cents per gallon. The total expenditure of operating the engine is 37 cents per hour, which compares with considerable over $2.00 per hour for engines operated by distillate or other high volatile fuels.

WASHINGTON-ESTEP MARINE DIESEL ENGINES

Four-Cycle, Full Diesel Direct Injection, 350 lbs. Compression.

8¾"×12" at 350 R.P.M. = 23.9 B.H.P. per Cylinder.
137% Stroke ratio 700 feet Piston Speed.
Approximate weight 200 lbs. per H.P.
2 Cylinder 45 B. H. P., Weight 9,000 lbs.
3 Cylinder 70 B. H. P., Weight 13,400 lbs.
4 Cylinder 90 B. H. P., Weight 18,000 lbs.
6 Cylinder 135 B. H. P., Weight 27,000 lbs.
5-inch Crankshaft.
10¾"×16" at 280 R. P. M. = 38.5 B. H. P. per Cylinder.
145% Stroke ratio, 746 feet Piston Speed.
Approximate Weight 230 lbs. per H.P.
3 Cylinder 110 B. H. P., Weight 25,200 lbs.
4 Cylinder 150 B. H. P., Weight 34,500 lbs.
*6 Cylinder 230 B. H. P., Weight 53,000 lbs.
6¼-inch Crankshaft.

(*Direct Reversible Type.)

16½"×24" at 210 R. P. M. = 102 B. H. P. per Cylinder.
149% Stroke ratio, 840 Piston Speed.
Approximate Weight 230 lbs. per B.H.P.
*3 Cylinler 300 B. H. P., Weight 69,000 lbs.
*4 Cylinder 400 B. H. P., Weight 92,000 lbs.
*6 Cylinder 600 B. H. P., Weight 138,000 lbs.
9¾-inch Crankshaft.

(*Direct Reversible Type.)

DIESEL ENGINES FOR SUBMERSIBLE CRAFTS.

That the Diesel engine has been found to answer the requirements necessary as a prime mover suitable for naval duties on submersible crafts has been fully proven. For many years the gasoline driven engine was the best at our disposal, but as gasoline is a bad thing to handle in the confined space of a submarine, the Diesel engine rapidly substituted the gasoline engine as soon as the advantages were known to governments throughout the world. The development of these engines was quite advanced in Germany before any such marine engines were built in this country. In order that we might advance as rapidly as possible, all known engines of this type were examined by our engineers, and the conclusion reached that the engine built in Nuremburg was the best then developed. Steps were immediately taken to acquire the rights for this country, and we were thus able to get for our submarines the best engine then developed. Many of these engines were built and are now in operation in our submarines. In the building and operation of these engines many things were found unsuitable for service in the United States, the principal reason was their complication of mechanism. In consequence a new engine was designed—illustrated in Fig. 1. Other models also were soon brought before the Bureau of Navigation, and it must be acknowledged that the United States Government has much stimulated and encouraged the development of Diesel machinery. Engines built for submarine service must be rigid in construction, excellent in workmanship, reliable in action, capable to stand every known abuse, and above all, accessible for inspection. The change from gasoline to heavy oil has brought out one very interesting characteristic, that is, that with a given quantity of heavy oil, twice the number of horsepower-hours may be obtained as from a like quantity of gasoline. Thus, with a boat having a given full tank capacity, double the radius of action is obtained when the change from gasoline to oil is made. Another point is that heavy oil costs about one-fifth as much per gallon as gasoline; thus for a given number of horsepower-hours the fuel of the Diesel engine costs about one-tenth that for the gasoline engine.

Figure 2 gives a good idea of the mass of equipment of a submarine, every part of the space being utilized. The picture is taken from amidship looking forward. In the center of the picture is shown the hand steering wheel. In general the steering is done by electric motor, shown at the top of the picture. On the left is the air manifold, with valves for control of the high-pressure air. These valves connect the air to all the different tanks. By opening the valves to the main ballast tanks the water may be blown out in a short period of time. On the right is shown the water manifold which connects the different tanks to the adjusting pumps, also the levers of the large Kingston valves. Figure 3 gives a view looking aft from amidships and showing the main motors and engines.

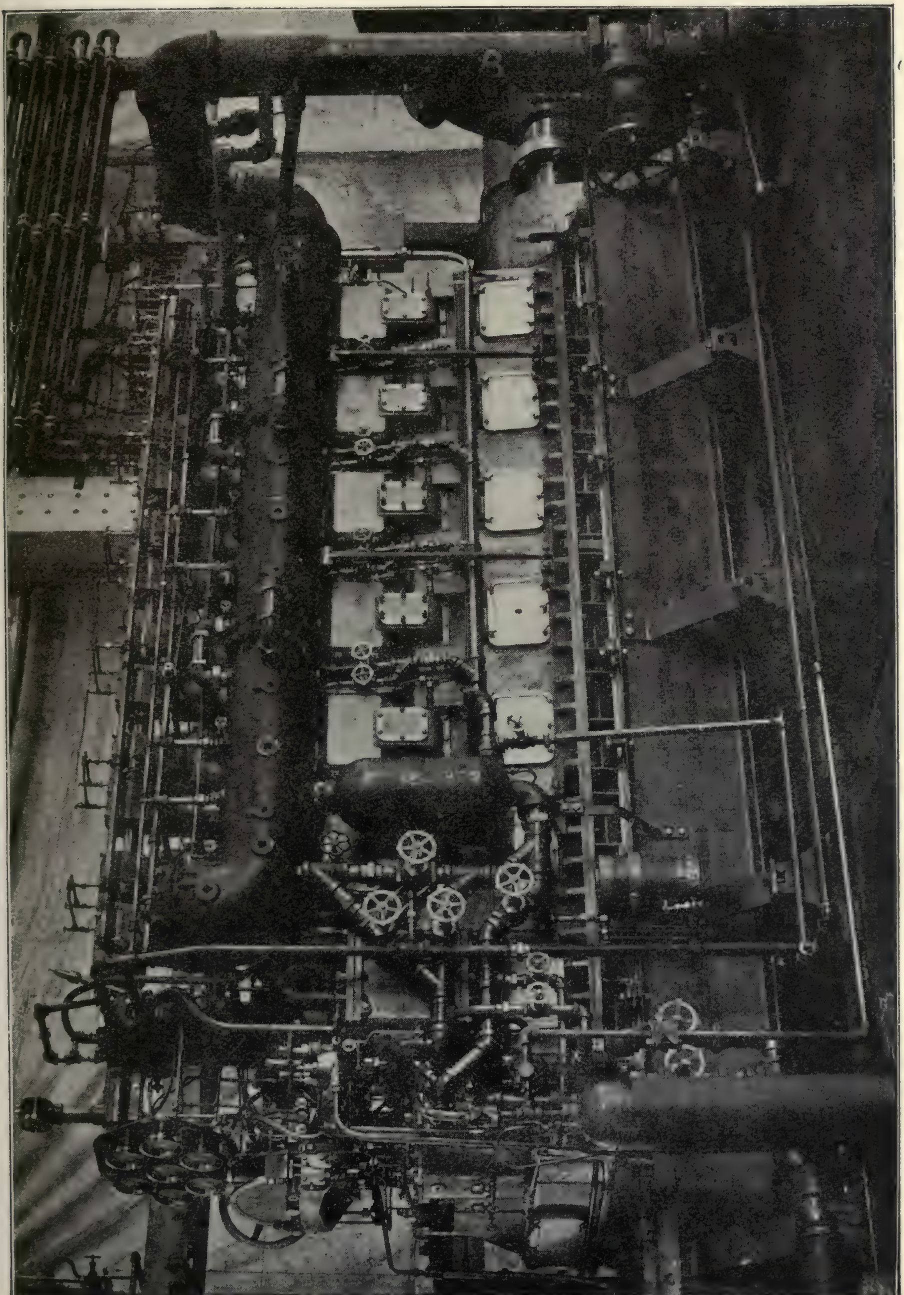


Fig. 1—Side View of 450 H. P. Two-Cycle Diesel Engine. The Two-Cycle as Well as Four-Cycle Have Proven Efficient in Engine Performances in Submarine Service

It becomes necessary here to mention the Automatic Blow Valve. This valve connects the high-pressure air line with the main ballast tanks, and the control of the valve is by diaphragm in connection with the outside sea water. Thus, if the pressure reaches too high a figure, the high-pressure air is automatically turned into the main ballast tanks. These tanks are entirely filled with water, whenever any is there, and therefore, at such times the main Kingston valves are left open. The turning of the high-pressure air into these tanks is all the operation required to empty the tanks. In the test the automatic blow valve is set to some depth, say 50 ft., and the boat allowed slowly to sink. When this depth is reached the pressure outside operates the valve and some 75 tons of water are quickly blown out of the tanks. The boat immediately starts to rise, and in less than one minute will reach the surface, nearly jumping out of the water from the rapid rise. The automatic blow valve may be set for any depth that may be desired.

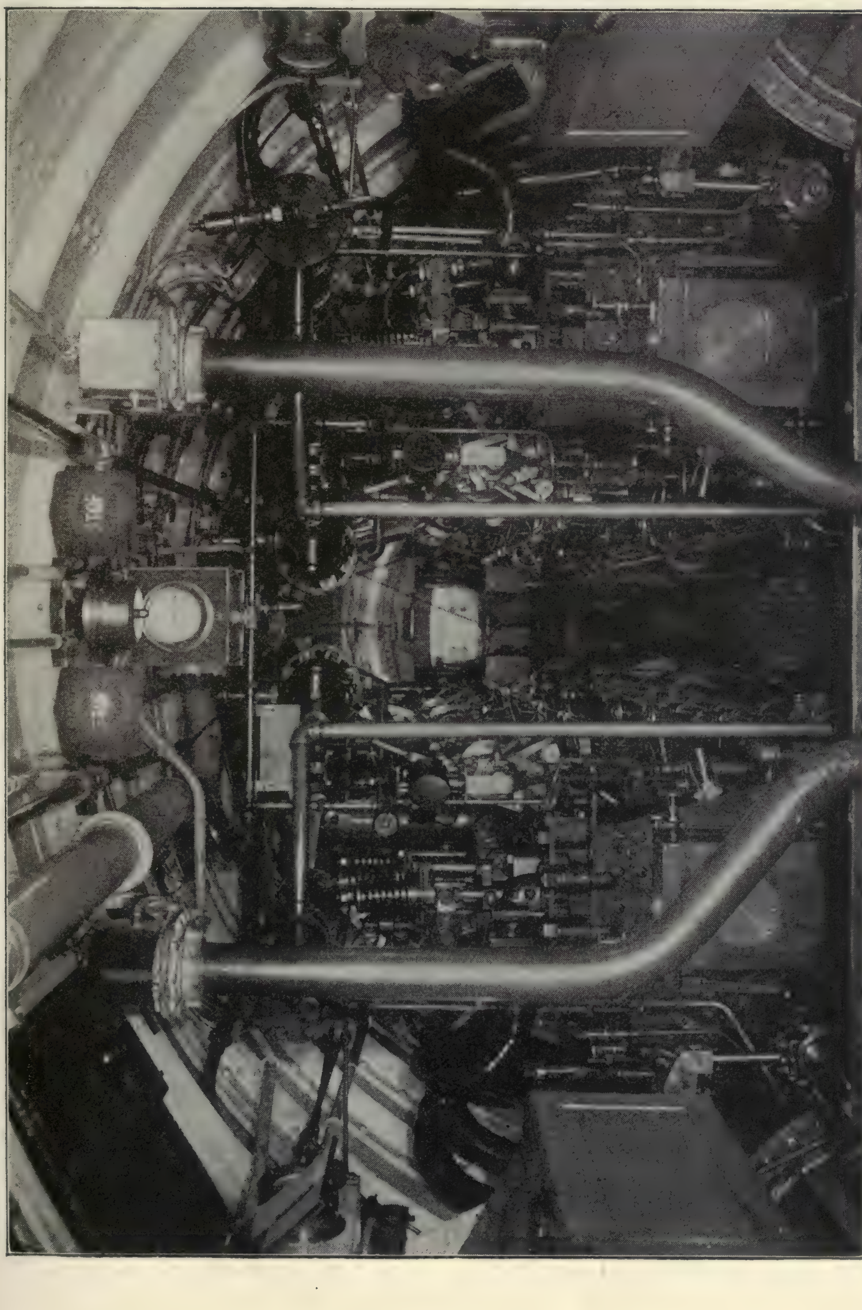


Fig. 2—From Amidship Looking Forward

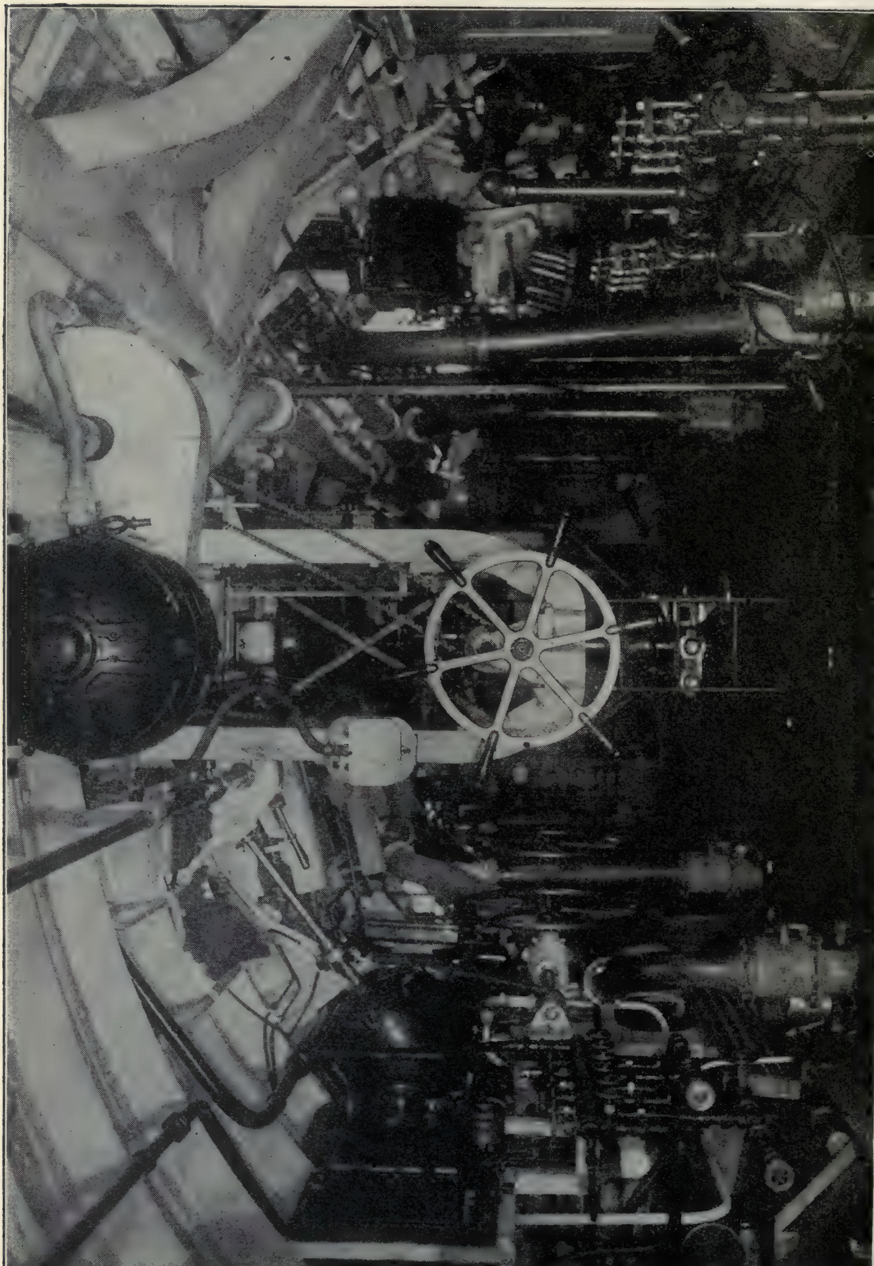


Fig 3—From Amidship Looking Aft

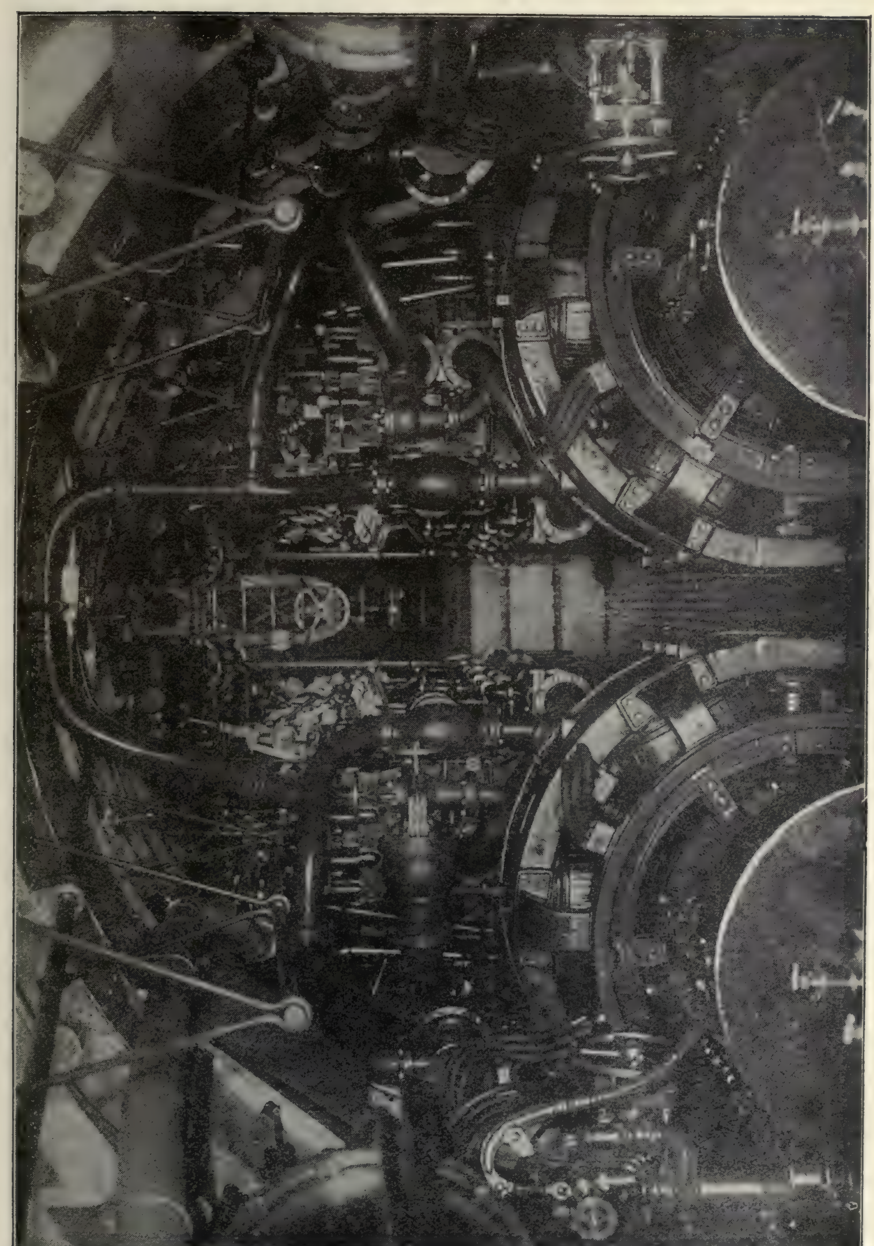


Fig. 4—From Aft End Looking Forward

Fig. 5—U. S. Submarine "K-1". Regular Performances Have Been Notable by Diesel Propulsion in This Service

CHAPTER XI.

DIESEL ELECTRIC PROPULSION.

*DETAILED INFORMATION AND OPERATING INSTRUCTION OF WESTINGHOUSE ELECTRIC MFGR. CO.'S DIESEL-ELECTRIC SYSTEM.

Diesel Electric System for Ship Propulsion: Although the Diesel electric system of ship propulsion is relatively new, the constituent parts making up the system are well established. A Diesel electric system preferably consists of two or more Diesel engine driven generators furnishing power to a motor driven propeller. The ship may be of the single, twin or multiple screw type. By using two or more generating units for a propeller, definite advantages in the way of weight, flexibility, control, reliability, etc., as discussed below, are readily obtained. The simplicity of the Diesel electric system is obvious when it is realized that the principal component parts comprise only four pieces of apparatus, such as Diesel engines, generators, motors and control, two of which are quite similar.

Selection of Power: In selecting the power for a Diesel electric system, we have a choice between alternating current and direct current. For the reason that direct current obviates operating difficulties ensuing from alternating current parallel operation; eliminates changes in engine speed; adds enormously to the simplicity, refinement and economy of control; and provides greater power in case of casualty to a generating unit, the direct current system is obviously the proper system to use. In cases where a single generator supplies power to a single motor, alternating current could be used without encountering difficulties ensuing from parallel operation, but such a system would be far inferior to the direct current system in flexibility, reserve power and control. Furthermore, a multiplicity of generating units, as are used in the case of D. C. systems, adds considerably to the ultimate reliability. Therefore, in the large majority of cases, Diesel electric systems will utilize direct current.

In the case of direct current systems, there are two general arrangements of machine connections that suggest themselves. One arrangement is to operate the generators in parallel, and control the motor speed and maneuvering by armature rheostatic means. This system,

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*Specially Prepared by W. E. Thau, Marine Engineer.

however, is rather cumbersome, wasteful during maneuvers and speed changes, and necessitates a complicated controller. The other employs what is known as the voltage control, or Ward Leonard control system. With this system, pure shunt machines are used and both motors and generators are separately excited, preferably from the same source. The motor fields are excited at constant potential, and always in the same direction. The excitation of the generator fields is varied to suit the motor speed and direction of rotation desired. By varying the voltage applied to the armature terminals of a shunt motor, having a constant field excitation, the motor speed can be varied in direct proportion, both as regards speed value and speed direction; and since the voltage generated by a constant speed, separately excited, shunt wound generator is directly proportional to its field excitation (neglecting saturation), the motor speed is, in turn, proportional to the generator excitation. With such an arrangement, therefore, it is only required to vary the generator fields from full excitation in one direction to full excitation in the opposite direction, to cause the motor to maneuver from full speed ahead to full speed astern. To further simplify this method of control, all machines are connected in series. With the series connection, it is unnecessary to maintain like speeds on all the engines. Provided the generators are excited equal amounts and have identical performance, the only effect of difference in engine speeds is a proportional difference in the loads carried by the generators, and their driving engines. From an operating standpoint, therefore, the series system is ideal, and permits by far the simplest system. Parallel operation of generators with the Ward-Leonard system would be very difficult, in fact, almost impractical.

Since it is only necessary to handle the generator field excitation currents for maneuvering the ship from full speed ahead to full speed astern, or holding any particular desired speed, the economy of the Ward-Leonard system is obviously superior to that of the armature rheostatic system during any other than full speed operation, for the reason that the generator field excitation power does not exceed 1½% of the total output of the generator. Dealing with these small currents, the control is extremely simple and inexpensive. This simplicity has a further direct effect on the maintenance of the equipment.

BRIEF DESCRIPTION OF UNITS:

Engines: The engine used with Diesel electric propulsion may be any reliable make of Diesel engine, which operates at a reasonably high rotative speed. The term "rotative speed" is used instead of "speed" to distinguish from high piston speeds. Many people associate the engines used with Diesel electric propulsion with those used for submarine propulsion. Diesel engines which are properly designed for use with Diesel electric propulsion need not exceed established safe piston speeds for continuous operating engines. By using many cylinders of

short stroke and small bore, the heat stresses common to large cylinder, slow speed engines are minimized, and the result should be an engine requiring less maintenance, and an engine of simpler construction.

By resorting to higher rotative speed engines, it is well known that the weight per brake horsepower can be brought down very rapidly. This characteristic is an important one in connection with Diesel electric drive, as the amount of weight thus saved in the engine is considerably more than that added by the electrical machinery, and hence results in a total machinery installation which is lighter than that of any other type. It is confidently anticipated that the weight of a Diesel electric propulsive installation using properly designed engines, should be approximately ½ that of a twin screw direct connected Diesel propulsive system, and in the neighborhood of 75% of the weight of an economical, geared-turbine propulsive equipment.

Generators: The generators used with the preferable form of Diesel electric propulsion are simple, direct current, shunt machines, the construction and performance of which are easily comprehended by any person having a mechanical turn of mind. These machines consist essentially of two parts, the field or stationary element, and the armature or rotating element.

The field is made up of a cylindrical steel ring, split at the horizontal center line for convenience, and having an elliptical section. Electrically, this frame serves to carry the field flux, and mechanically to support the field poles. The frame is machined on the inside diameter in order to form a true seat for the field poles which are bolted to it and symmetrically spaced. The main field poles are composed of a number of die-punched laminations of sheet iron, which are riveted together to form a solid pole. The commutating field poles are built of solid steel and located between the main poles.

The main field coils which produce the field flux consist of a large number of turns of insulated copper wire having a relatively small section. The coil is wound on a form, slipped on the field poles before they are bolted to the frame, and rigidly supported from these field poles by insulated supports. These coils are known as shunt coils, and are connected in series.

The winding for the commutating field pole consists of a relatively small number of turns of bare copper strap secured by insulated supports. This winding is connected in series with the armature, and carries the line current. The purpose of the commutating pole winding is to provide a magnetic field to neutralize the effect of the current reversal in the armature coils undergoing commutation, and thus to effect sparkless commutation. Since the commutating field winding is in series with the armature, and carries the same current, the correct amount of commutating pole flux is automatically provided under all conditions of load within the capacity of the machine.

The armature consists essentially of a cylindrical core built up of steel laminations which are dovetailed and secured to a cast spider, the

spider in turn being pressed and keyed on to the shaft. The steel laminations are provided with teeth punched in their periphery, and into which the armature coils are placed.

The commutator to which the armature coils are connected is made up of a series of hard drawn copper bars securely insulated from one another by means of mica insulation. These commutator bars are built up on a separate spider and securely fastened thereto by means of "V" rings fitting into insulated machined recesses in the bars, or by some other suitable means. The commutator spider is then pressed on an extension of the armature spider, or directly on the shaft and keyed thereto.

The armature coils are form wound and completely insulated and treated so as to be moisture resistant, before they are placed in the slots, and connected to their respective commutator bars. The armature coils for any given machine are identical.

The armature is usually carried on a forged steel shaft having an integral flange at one end which is bolted directly to the flywheel of the engine, the other end being carried by a pedestal type bearing.

The brush rigging which properly constitutes a part of the stationary member, serves to collect the current from the commutator, and is supported from the field frame. There are the same number of brush arms as main field poles. Brush arms are symmetrically placed and so located that the brushes rest on commutator bars which connect to armature coils, which lie in the commutating zone, which in a commutating pole machine is midway between the main field poles.

The brush arms carry a series of brush holders, each of which is provided with a carbon brush connected to the brush holder by means of a copper shunt (sometimes called a pig-tail). To insure an equal distribution of current in the brushes, an adjustable spring is provided on each brush holder which maintains the pressure of the carbon brush on the commutator at a given predetermined correct value.

Motors: The corresponding description of the motor is identically the same as that of the generator, and therefore, will not be given. In order to minimize the total weight, it is preferable to provide the motor without a bedplate, and simply to provide feet on the field frames and bearing pedestals suitable for mounting on a built-up structural steel bedplate in the ship. The structure supporting the motor should be rigid so as to avoid distortion.

The bearings of the generators are usually supplied with lubrication from the engine lubricating system. In the case of the motors, it is usual to provide oil ring lubrication. In some cases, however, forced or flood lubrication is provided, the oil being supplied by a gear pump actuated from the motor shaft.

Exciter Arrangements: The exciters for the generator and motor fields may either be driven by the main engines, or by separate engines. In either case, they may furnish power to the auxiliaries in addition to that for excitation.

If driven by the main engines, the exciters should be direct connected to the main generator either by means of a coupling, or by mounting on an extension of the generator shaft. In the latter arrangement, the exciter armature may be overhung if the mechanical factors permit. Driving the exciters from the main generator shafts by means of chains, belts or gears, effects a slight saving in weight and overall length of the set, however, it is not nearly as satisfactory mechanically as direct drive.

Whether it is best to use direct driven, or separately driven exciters, depends upon such factors as sea load, desired flexibility, capacity of main generators as related to port demands, available space, etc., and each case must be considered on its merits. When the arrangement is convenient, it is usually preferable, however, to drive the exciters by the main engines, as it results in a self-contained propulsive plant.

Control: The control consists of a suitable switchboard containing the necessary switches for the several machines involved, the instruments, protective relays, circuit breaker, etc.; a special reversing field rheostat for the generator field circuits; and a manually operated, remote control mechaniam, preferably mounted on a pedestal for operating the field rheostat.

There are two general methods of operating the field rheostat. One method employs a handle which operates in the fore and aft direction, and is thus similar to present steam engine control. This handle operates the rheostat through a system of rods and bevel gears. The other system employs a worm and wheel instead of the handle, and is preferable to the handle for the reason that it provides an inherent time element in that it requires a certain definite time to actuate it from full ahead to stop, and full astern positions. This time element is essential, as too rapid change in the field strength would cause serious overloads on the machinery. It has been found from actual service that the minimum time which would be consumed in bringing the propellers from full speed ahead to the stopped condition is approximately 5 seconds, and by designing the worm and wheel so that it would require 5 seconds to make the number of turns necessary for full speed to stop position, this required time element is automatically provided. With the lever operation, it is possible to move the control instantly to the off position and thereby cause a large rush of current.

The switches for the machines are so arranged that any particular generator or motor unit may be taken out of service by simply throwing its switch from one position to another. This operation is usually effected without interrupting the service to the propeller motor.

A more detailed description of the control is given below in the case of a specific example.

ADVANTAGES:

The advantages of Diesel electric propulsion as compared with any form of steam drive are very pronounced.

Fuel Economy: The most important advantage is that resulting from the difference of fuel economy. A properly designed Diesel electric propulsive equipment requires about 0.55 lbs. of oil per shaft horsepower hour for all purposes, whereas the average high grade steam installation requires about twice this amount, or more. The saving in fuel is of two-fold importance. First, the actual difference in fuel cost, and second, the additional cargo that can be carried due to the decreased weight of fuel, or the greater cruising radius for a given amount of fuel oil.

Weight: A proper Diesel electric propulsive equipment being lighter than the geared turbine equipment of high grade performance, enables additional cargo to be carried in the amount of the weight difference. The importance of this feature is apparent as it affects the ultimate earning ability of the ship.

As compared with a direct connected twin screw Diesel drive, the Diesel electric is about on a parity in regard to fuel consumption, but considerably lighter in weight.

The direct drive Diesel has demonstrated its superior economy in the earning power of the ship as compared with the steam drive. The result was accomplished in spite of the excess machinery weight of the former as compared with the latter. Similarly, as the Diesel electric is considerably lighter than the direct drive Diesel, the Diesel electric will show superior earning power as compared with the direct drive Diesel, particularly on long runs.

Reliability and Reserve Power: By providing a number of small generating units, the reliability of the Diesel electric drive is superior to that of the direct connected Diesel drive, both from the standpoint of individual engines, and the drive collectively, in the case of casualty. This superiority of the Diesel electric also obtains when compared with the steam drive.

Reserve power in case of casualty to any of the generating units is important. Having a number of generating units, the reliability in case of casualty is infinitely greater than in the case of a single screw steamship. Although with a cross compound geared turbine, the failure of one element would still enable operation at about 50% power from the remaining element, it would be at a sacrifice of considerable speed and economy. A similar analysis applies in the case of the twin screw direct connected Diesel. To provide a motor of small diameter it is customary to use what is known as a double unit motor which, incidentally, results in greater reserve power flexibility. The double unit motor essentially consists of two electrically independent motors mounted on the same shaft. With such an arrangement, the reliability and reserve power of the Diesel electric in case of casualty, is infinitely superior to that of the single turbine and single direct drive Diesel single-screw ship.

With the Ward-Leonard system, using the series arrangement of machines, more reserve power is available in case of casualty to a prime mover than is the case with any other system of ship propulsion. Taking, for example, a 3-generator, single-screw arrangement, the failure of one generating unit would enable 88% speed to be obtained with the remaining two sets, and in the case of the failure of two generating sets, the single remaining set would furnish sufficient power to propel the ship at 70% speed. This analysis is based on the driving power varying as the cube of the speed. This system is in fact the only system that permits full power to be derived from remaining units without overloading them, or increasing the original size and weight.

To obtain the full capacity of the remaining units under conditions stated in the foregoing paragraph, it is merely necessary to set each generator for its full rated voltage, and then to decrease the motor field current until full load armature current flows through the system. In the example cited, full load current with two generators in operation would supply two-thirds of the total power, and with one generator in operation, would supply one-third full power. It is necessary to weaken the motor field to obtain the required speed for the remaining power, as otherwise the motor would operate at a speed directly proportional to the total remaining generator voltage. If the motor is of the single armature type, its field flux would be reduced to 76% to obtain 88% speed with two generators in operation; and 47½% with one generator in operation. If the motor is of the double unit type with its armatures normally operated in series, 70% speed can be obtained with one generator, and one motor unit, and in this case this motor field flux is reduced to about 95% of full value.

In the case of a twin or multiple screw ship having Diesel electric propulsion, a casualty to a prime mover does not prevent supplying balanced power to all screws. The switching is so arranged that the generators may be connected to any of the motors. This is of further advantage in that the remaining prime movers are operated under normal power conditions and consequently normal efficiency.

Recalling the reliability which was discussed above, it is inconceivable to imagine a reasonable condition of casualty in the case of Diesel electric propulsion that would prevent the ship from reaching port at a reasonable speed.

Simplicity: The characteristics of the engine for Diesel electric drive are constant speed and reasonably close regulation from no load to full load. Since the engines operate at constant speed and always in one direction, it is unnecessary to point out the elimination of the reversing gear, as well as the air for reversing. The result of the elimination of these two features means a simpler and probably a more reliable engine. It at least reduces the air problems to their simplest terms. Air is used only for the initial start in port. In fact, the plant can be arranged so that only one engine need be started by air and subsequent engines started electrically by utilizing their generators as motors.

Furthermore, as an extreme arrangement, starting air may be eliminated entirely by providing a small gasoline or kerosene engine generator set to do the starting of the main engines electrically.

Stand-by Conditions: Since the Diesel engine consumes fuel only when running, a further economy is effected by the elimination of stand-by losses. Also, it is unnecessary to warm up various pieces of machinery, such as boilers, turbines, etc., for a long period prior to "getting under way." The Diesel electric system can be made ready for sailing on short notice.

Conclusion: From the foregoing discussion, it will be obvious that the Diesel electric system of ship propulsion using series connected D. C. machines operating on the Ward-Leonard principle, is considerably more than a mere electric coupling or gear. It is a system containing very pronounced features which are of direct advantage to the improved performance of the ship. The fact that the electrical machines constitute a reliable and flexible substitution for gears and magnetic couplings is merely incidental.

LIMITS OF CAPACITY:

Based on present-day available Diesel engines, which are suitable for Diesel electric drive, the capacity limit of a single Diesel electric drive, is approximately 7500 H. P. This figure could be increased by using an unreasonably large number of engines. Some ultra-enthusiastic advocates of Diesel electric propulsion have entertained the idea of using as many as 18 engines. However, the more conservative advocates would limit the number of engines for a single installation to eight, and this only as an extreme measure. The preferable number of engines for a single-screw drive is three or four.

Engines of 1000 H. P. per cylinder are now being seriously considered, and in fact, developments of such engines are already under way. Using six or eight cylinder engines, having this capacity of cylinder, it is easily conceivable that single installations of 50,000 H. P. are on the horizon.

With possible future developments in Diesel engines, the limit may be within the greatest demand for a single drive. As Kipling said, "Came the power with the need."

PERFORMANCE:

Stopping and Reversing: Because of the inherent functioning of a Ward-Leonard system of Diesel electric propulsion it is necessary to give some thought to the inherent characteristics of propeller performance, particularly quick stopping when under full headway. Turning warrants consideration only in the case of multiple screw ships, and only then in particular types of drive, such as turbine-electric using alternating current machinery.

During quick stopping, however, it is necessary to overcome the propeller torque in order to bring the propeller to rest. This propeller torque is developed by reason of the motion of the ship through the water, which causes the propeller to be driven as a water motor. Although strict analysis of this performance is not pertinent to the present discussion, it is well to recognize the results. In the case of a pure Ward-Leonard system of Diesel electric propulsion, the principal means of absorbing the energy returned through the screw is the friction of the engines. If more energy is returned than can be dissipated by the friction losses in the engines, and the losses of transmission, external means such as dynamic braking resistors must be provided. If all the energy returned by the screw is not dissipated by the frictional losses of the engines, or otherwise, it will be expended in increasing the speed of the engines. Whether or not the increased speed would be detrimental to the engines can only be conjectured. However, it is confidently believed that in the large majority of cases, and particularly those of the ordinary cargo ship, that this energy will not be in excess of that which can be obsorbed by the frictional losses of the engine. Specific tests of this performance were made in the case of some Diesel electric yachts when stopping from full speed, and it was found that there were no increases in the engine speeds due to this cause.

A conservative analysis shows that the maximum horsepower returned to the engines when bringing the propellers to rest from full speed, is approximately 33%, and that this peak value will last for a very brief instant only. The average horsepower returned is approximately 18%. The values take into account the torque produced by the propeller and the inertia forces of the motor armatures and the propeller, based on a stop of five seconds. Since the average four-cycle engine with attached auxiliaries is approximately 75% efficient, and the average two-cycle engine with attached auxiliaries is approximately 72% efficient, based on brake horsepower, it is apparent that the frictional losses in either case amount to at least 33% of the engine output. There is a slight margin in these figures, as no allowance was made for the propeller shaft bearing friction. Due to this returned energy when making quick stops, or reversals, it is necessary to design the engine governors to throttle to practically zero oil flow, in order that the friction losses of the engine may provide a load for absorbing the returned energy.

In cases where the compressors, etc., are separately driven, and in cases where the returned energy otherwise is in excess of what the engines can absorb, it is necessary to connect resistance in the circuit during quick stops, for absorbing the excess. This is very easily arranged, and its insertion is done automatically by an auxiliary circuit actuated by virtue of relative position or motion of the control device.

Turning: When turning at full power, there is an increase in the load on the propellers. This increase is particularly pronounced in the case of multiple screw ships, as the power builds up enormously, especially on the inboard side of the turn if means are not taken to guard

against it. This characteristic causes some concern in the case of alternating current drives, and necessitates special control devices. Even though the input to the prime mover be limited to normal running value, there is a drop in its speed and an increase in torque of approximately 25% to 30%. Since a normal Diesel engine has very little overload capacity, any building up of torque will cause a reduction in its speed, and therefore, its output is automatically limited. The inherent characteristics of the D. C. motors and generators are such that they will carry the increase in torque without the least danger of becoming unstable, and hence no special precautions need be taken. There is always a stable couple between the generators and the motors.

Bridge Control: Since the engines operate at practically constant speed, and in the same direction at all times, and are under the control of a constant speed governor, they require no attention during maneuvering. This combined with the absence of such factors as steam pressure, boiler fires, priming, etc., make control of the propeller machinery from a remote location entirely feasible. In other words, the ship is controlled just as easily from the bridge as from the engine room. The importance of such performance is obvious in the case of ships requiring very accurate maneuvering in restricted places, as it eliminates both delay in response to signals, and risk as a consequence of mistaken signals. The Diesel electric system is, in fact, the only system of ship propulsion that affords bridge control without resorting to complications which are questionable at best.

Torque at Low Speed: In regard to low speed torque, the Diesel electric system here described is very similar to the turbine, in that it is capable of developing large overload torque at reduced speed: Theoretically, the torque at the motor shaft may be increased in inverse ratio to the speed without overloading the engines. However, safe commutation of the electrical machines of ordinary design places a practical limit on the torque that can be developed under these conditions. For the purpose of ship drive, however, the ordinary direct current machine will develop sufficient torque to meet any emergency.

Special Provisions: A few special precautions are necessary. Overload protection is necessary to prevent serious injury to the electrical machinery. This protection is provided in the form of a circuit breaker whose function it is to open the circuit on excessive overloads. Owing to the desirability of maintaining continual control of the screw under all conditions, the adjustment of the circuit breaker is such that it will not open under any normal operating conditions of the ship. The idea in providing the circuit breaker is merely to provide protection to the machinery in the case of an equivalent of a short-circuit.

Assuming that the circuit breaker has tripped while the ship is under way, and it is desirable to immediately regain control of the screw, it is necessary to first establish proper voltage conditions on both sides of the circuit breaker before the circuit breaker can be closed.

The motor will generate a counter voltage due to the fact that it is being rotated by the action of the propeller. To close the circuit breaker without jarring the machinery or producing a large rush of current, it is necessary that the generator voltage be somewhere near that of the motor voltage. To effect the closing of the circuit breaker at the proper instant, voltage balance relays are provided, the function of which is to prevent the automatic circuit breaker from closing until the voltages on both sides of its contacts are approximately the same. This device is automatic and is described below in the case of a specific example.

In the case of a number of generators connected in series, the failure of power on one of the engines would result in that generator stopping, reversing and speeding up in the opposite direction as a motor, being supplied with power from the remaining generating sets. The speed which this generator would obtain depends upon the total amount of voltage it would absorb from the system, and if the number of generators on the system is sufficient, the speed might be excessive, and would likely result in casualty to the engine. To prevent this, some device which will automatically trip the field of the inactive generator, or perform an equivalent service, must be provided. There are several ways of accomplishing this result.

In the case of double-ended ferry boats using the Ward-Leonard system of control, it is very likely that the energy returned during quick stopping or reversal from full speed would be more than could be taken care of by the losses in the engines, due to the fact that there are two screws returning energy instead of one. To prevent the return of energy from both screws, it is necessary to either provide a dynamic braking resistance to absorb the excess over what the engines can take care of, or to make one motor inactive during this period. To make one motor ineffective, a system has been devised which utilizes a current relay. During the stopping period, this relay regulates the current in one of the motor circuits to a very small value, and thus prevents that particular motor from returning energy.

APPLICATIONS:

Because of the advantages in fuel economy, weight, control, reliability, flexibility, etc., of the Diesel electric system of propulsion, it is largely applicable to cargo ships, coastwise vessels, liners, certain Naval craft, fishing boats, yachts, ferry boats, barges, lake boats, river boats, cable ships, fire boats; in fact, any ship where economy, refinement of control, good maneuvering characteristics, etc., are of any importance.

The following table indicates those advantages offered by Diesel electric drive which are of particular importance in the various types of ships:

PRINCIPAL APPLICATION ADVANTAGES

Main Machinery Only

Type of Ship	Fuel Consump.	Machy. Wt.	Refined Control	Reliability	Reserve Pr. & Flexibility	*Port Consumption	Cruising Radius
Cargo ships	X	X		X	X		X
Coastwise vessels		X		X	X		
Liners	X			X	X		
Certain Naval craft	X	X	X	X	X	X	X
Fishing boats	X			X	X		
Yachts	X		X	X	X		X
Ferry boats	X		X	X	X	X	
Barges	X	X	X	X	X	X	
Lake boats	X	X		X	X	X	
River boats	X	X	X	X	X	X	
Light ships			X	X		X†	
Cable ships	X	X	X	X	X		X
Fire boats		X	X	X	X	X	
Self-propelled dredges	X	X	X	X		X	

* This refers principally to stand-by losses.

† Consumption while on duty.

A 2500 S. H. P. DIESEL ELECTRIC DRIVE.

General Description and Arrangement: To convey a clear idea of Diesel electric drive, it is thought best to describe a specific case. The example selected is a 2500 S. H. P., single screw drive, having four 500 KW., Diesel driven, 250 volt, generators supplying power to a 2500 H. P., double unit, 90 R. P. M. motor. Two 75 KW Diesel engine auxiliary generating sets are provided for supplying the excitation and auxiliary load while at sea. The following is the list of apparatus constituting the drive:

1—2500 H. P., 90 R. P. M., double unit, direct current, 500 volt, shunt motor. The two armatures are mounted on a forged, flanged shaft carried by two pedestal bearings. The motor frames and the bearing housings and bearings are split along the horizontal center line to provide easy access. Fig. 1 shows a view of a motor of this type on test. Figure 2 shows the field of a similar motor. Fig. 3 shows a view of a small D. C. propeller motor with thrust bearing and bedplate arranged for mounting on a wooden foundation.

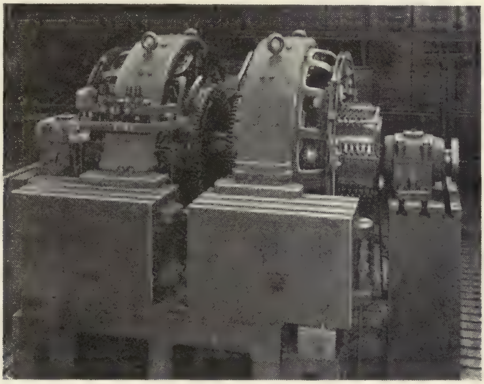


Fig. 1—Typical Double Unit Direct Current Shunt Propellor Motor

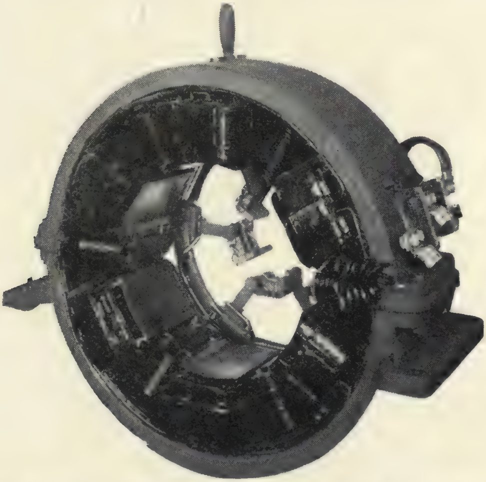


Fig. 2—Field of Typical Direct Current Shunt Motor or Generator

4—500 KW., 250 volt, direct current, shunt generators, direct coupled to four Diesel engines. The generator armature is mounted on a forged, flanged shaft supported at the commutator end by a pedestal bearing

and coupled to the engine flywheel at the rear end. As in the case of the motors, the frame and bearings are split. Fig. 4 shows a view of a Diesel engine generator set, the arrangement of which corresponds to that described.

2—75 KW., 250 volt, compound wound, direct current, Diesel engine driven, auxiliary generators. The mechanical arrangement is the same as that of the main generators. One of these sets serves as a spare.

1—Complement of motor driven, engine room, auxiliaries, such as oil pumps, circulating pumps, auxiliary air compressor, sanitary, fresh water, fire and bilge pumps, etc.

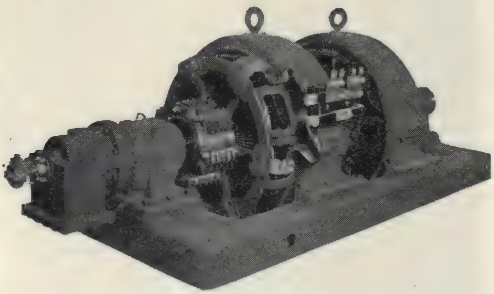


Fig. 3. Double Unit Propeller Motor with Bedplate and Self-Contained Thrust Bearing

1—Switchboard and control for the above machinery.

As the description of Diesel engines is well covered on other pages of this book, and as the electrical machines are briefly described previously in this chapter, no further description is given here. For a comprehensive treatise of the design, construction and characteristics of electrical apparatus, the reader is referred to any of the many reliable electrical text books found in libraries and book establishments. The control, its arrangement and operation are, however, briefly described below.

Fig. 5 shows the plan view of the machinery and its arrangement in the engine room. The four main Diesel generating sets are located forward; and the motor, auxiliary Diesel generating sets, vice bench, switchboard and control station are located aft. The oil supply tanks and other accessories are located on the upper grating (not shown). The location of the control station is such that the operator has full view of the propelling machinery, and hence the operator can observe the performance at all times. The engines are arranged right and left hand, so that their controls and gauge boards may be conveniently handled and observed. The entire arrangement provides accessibility and convenience for operation and inspection, and at the same time is not wasteful of space.

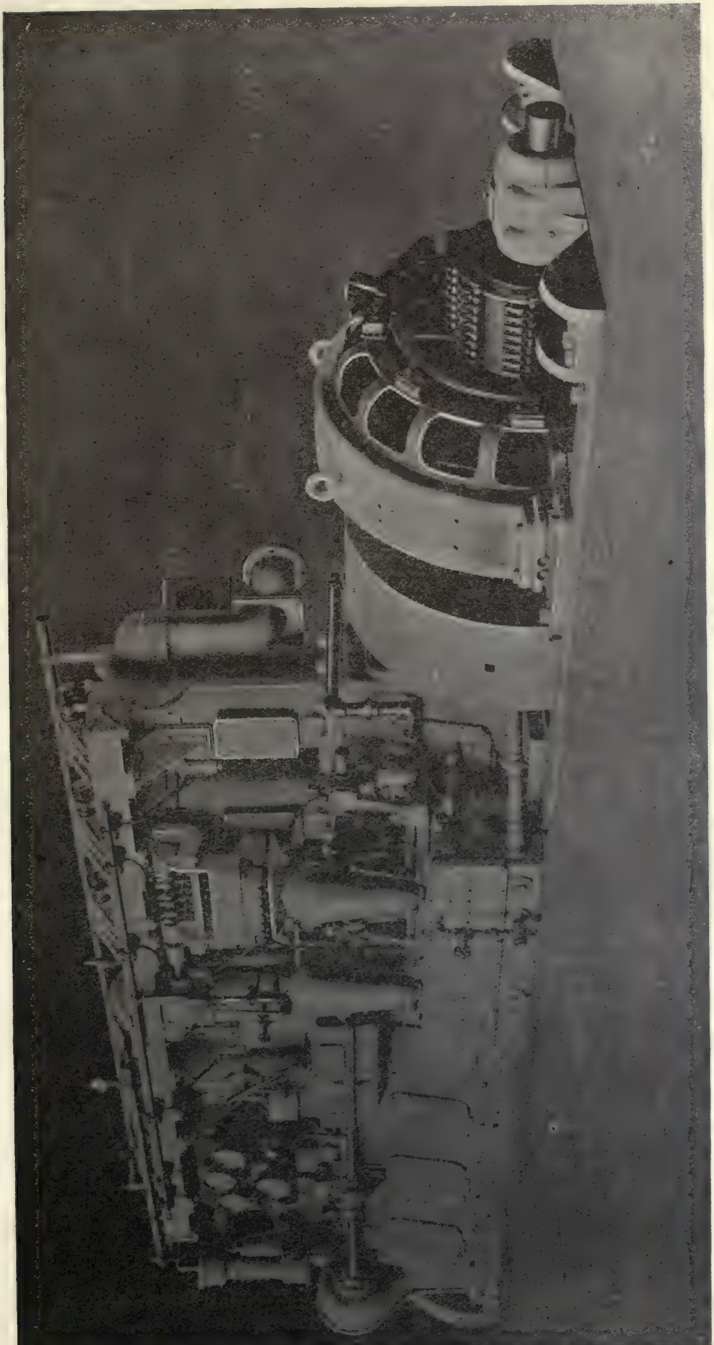


Fig. 4—Diesel Engine—Generator Set

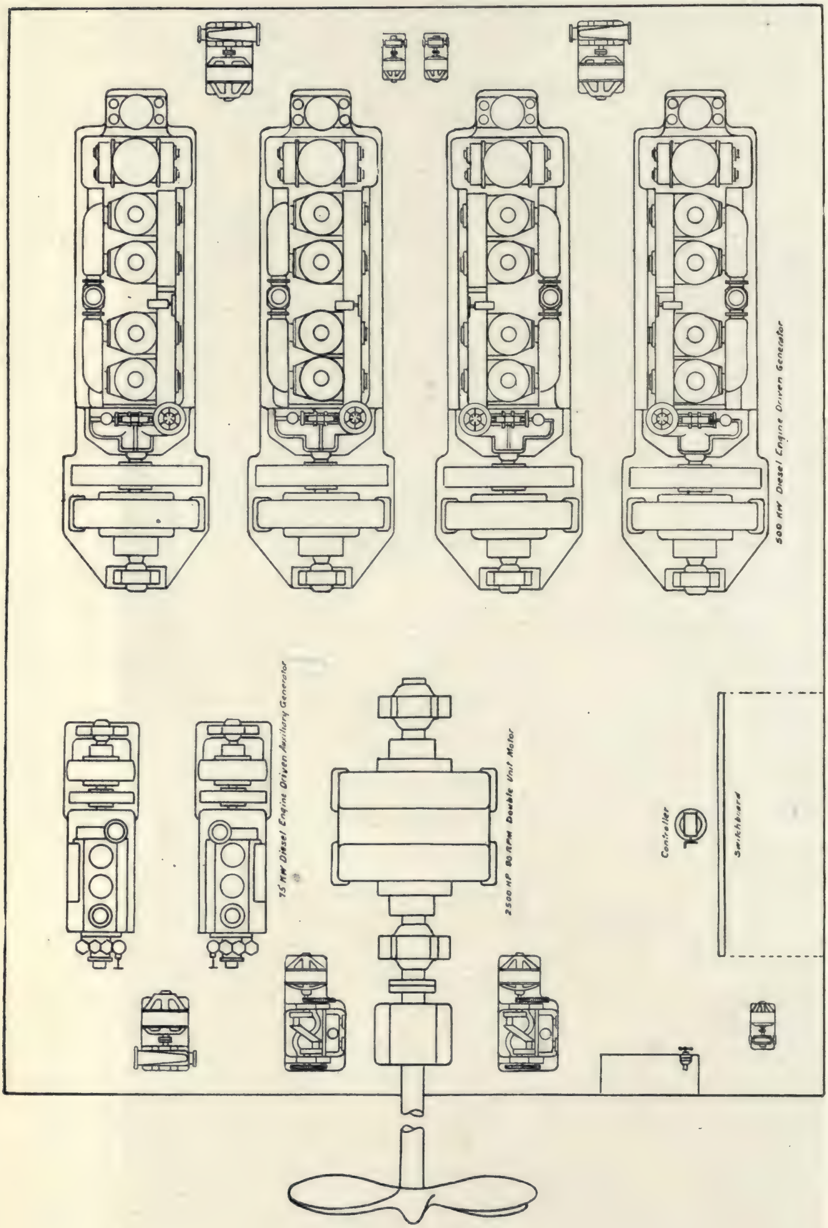


Fig. 5—Plan View of Engine Room for a 2500 S. H. P. Diesel Electric Drive

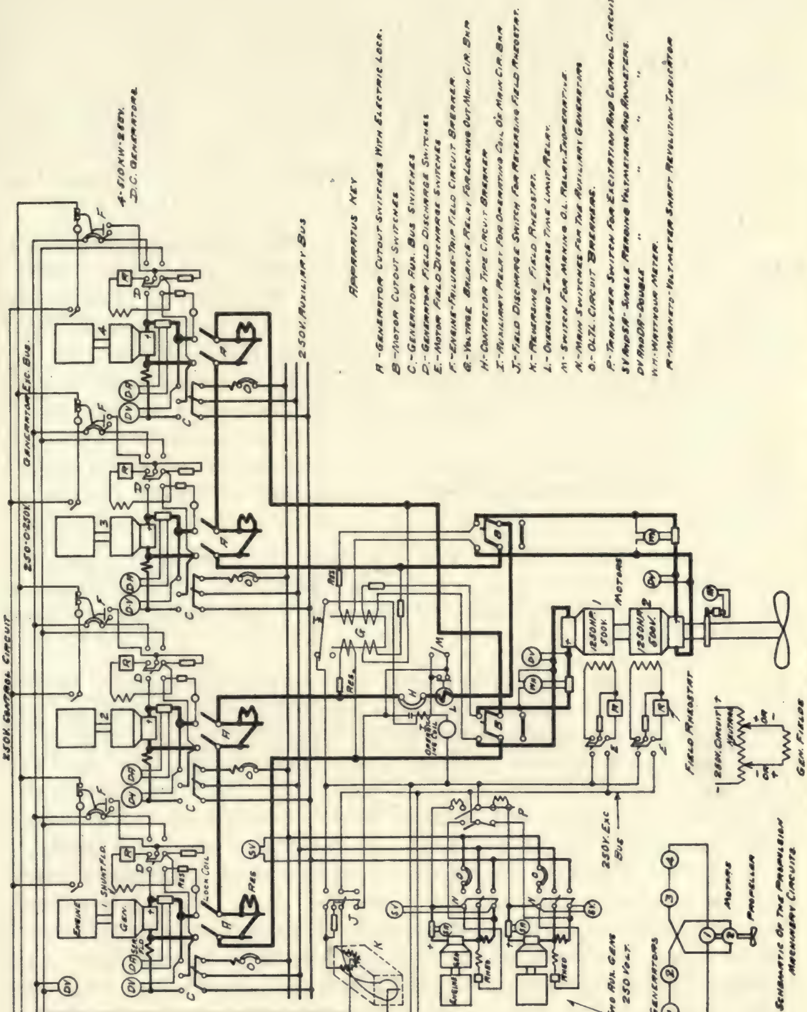


Fig. 6—Control Diagram for a 2500 S. H. P. Diesel Electric Drive

SWITCHBOARD AND CONTROL:

Diagram: The control diagram is shown in Fig. 6. The small diagram in the lower left hand corner shows the scheme of connections, and the main diagram shows the full details of connections, including all necessary switches, circuit breaker, relays, etc. A glance at the diagram will show that all main machines are connected in series, and furthermore, that the motor and generator armatures are interspersed so that the current passes through the circuit in the following order: Generator No. 1, Generator No. 2, Motor No. 2, Generator No. 3, Generator No. 4, Motor No. 1, and back to Generator No. 1, thus constituting a series circuit.

Since the total voltage of a chain of series connected generators is the sum of the voltages of the individual machines, the total effective voltage in this case is 4 $\times$ 250 or 1000 volts. However, by interspersing the motor armatures in the manner stated, the ground voltage, or the maximum voltage between any two points in the system is only 500 volts. Such an arrangement is advantageous in that the circuit is really a 1000-volt system from a current standpoint, but only a 500-volt system from a voltage or insulation standpoint. In other words, it necessitates only one-half the copper that would be required in a 500-volt system of this same capacity, and at the same time does not exceed the 500-volt insulation strain.

Switches, Relays, etc.: The principal switches, relays, etc., are designated by letters, and all switches performing the same function bear the same letter. For instance, all generator cutout switches are designated by the letter "A."

"A"—Two pole, manually operated, transfer, knife cutout switch for generators. When closed in the upper position, these switches connect their respective generators in the propulsion circuit; and when thrown to their full lower position, their blades connect to a solid bar, thus cutting the generator out of the propulsion circuit and establishing the series propulsion circuit through the bar between the lower contact jaws. The upper and lower portions of the blades are at an angle, so that the switch makes contact on the first set of lower jaws before breaking contact on the upper jaws, and vice versa. In the lower throw, the right hand lower blade engages two jaws in sequence. The first jaw inserts a resistance which prevents a rush of current, due to the residual field of the generator. Further closing breaks the connection to the generator armature on the upper jaws, and engages the bar on the lower jaws. This switch is electrically locked against being thrown to the lower position until the generator field has been opened.

"B"—Two pole, double throw, motor cutout, manually operated knife switch. This switch has no special features as it is not operated when the circuit is alive. The upper position connects the motor in the propulsion circuit, and the lower position cuts out the motor and establishes the propulsion circuit through the bar between the lower jaws.

"C"—Three pole, single throw, main generator auxiliary bus switches. These switches are provided in order to utilize the main generators when in port for supplying auxiliary power. It will be noted that this switch is three pole to permit parallel operation (equalizer connection), and that a series field is connected in the circuit to give the generator the desired compound characteristics.

Switches "A" and "C" are interlocked so that only one or the other can be closed in the upper position at the same time. This prevents using the generator for two purposes.

"D"—Generator field switches.

"E"—Motor field switches.

"F"—Engine failure trip, field circuit breaker. These are provided to automatically make the generator ineffective and to prevent its motorizing in the event of failure of its engine. This breaker is connected in the separate excitation circuit only as such protection is unnecessary when the generators are operating on the auxiliary bus. The means for opening this field circuit breaker is actuated by a mechanical attachment on the engine, or by a voltage differential relay; the latter, however, is rather complicated.

"G"—Voltage balance relay for insuring that the motor counter voltage and the generator voltage are approximately at the same value before the automatic main circuit breaker can be closed. This lock-out feature is necessary in case the main circuit breaker trips while the ship is under way, as explained above.

The device consists essentially of a spring-closed relay contact in the auxiliary circuit of the closing coil of the main automatic circuit breaker, and a polarized magnet, one pole of which is excited by the generator voltage and the other pole of which is excited by the motor voltage. When the two voltages are equal, the flux produced in the magnet core by the two windings is neutralized and there is no pull on the relay arm, and the auxiliary circuit to the main circuit breaker coil remains intact. If the generator voltage is appreciably different than the motor voltage, a pull is exerted on the relay arm by the polarized magnet, and the relay contact is opened, thereby preventing the main circuit breaker from closing.

"H"—Automatic reclosing circuit breaker located in the propulsion circuit. As stated above, the function of this device is to protect the machinery against practically short-circuit conditions. The breaker is provided with an inverse time element overload relay. In the event of very severe or sustained dangerous overloads, this relay, whose magnet is excited by the main current, will open the circuit of the circuit breaker closing coil through the auxiliary relay, and disrupt the main circuit. To again close the circuit breaker, it is necessary to adjust the generator voltage by means of the main control handle, to the value of the motor counter voltage. To re-establish the proper generator voltage, it is merely necessary to move the control handle slowly to the "increase" or "de-

crease" position, as the case may be, and when the proper position is reached, the breaker will close automatically.

"I"—Auxiliary relay through which the operating coil of the main circuit breaker is excited. This auxiliary relay is provided for the reason that the polarized relay contacts are of insufficient capacity to handle the current of the main circuit breaker closing coil.

"J"—Field discharge switch for the reversing rheostat.

"K"—Reversing field rheostat for main generator field excitation when generators are connected to the propulsion circuit.

(**Note.**—When the main generators are used for auxiliary power, they are self-excited and operate as normal compound wound generators.)

The simple diagrammatic scheme of the type of field rheostat used for the control of the propulsive machinery is shown in the lower left hand corner of the main diagram in Fig. 6. The rheostat is constantly energized from the excitation circuit. The leads to the field slide symmetrically over buttons on the rheostat face plate which are connected to the resistance at regular intervals. The arrangement is such that the lead contacts of the field circuit effectually cross each other at the middle point of the rheostat in going from full excitation in one direction to full excitation in the other, and hence not even a field circuit is opened in going from ahead to astern.

"L"—Overload inverse time limit relay. This is described under "H."

"M"—Switch for making overload relay inoperative. The purpose of this is to provide a means for making the circuit breaker non-automatic in the event that maneuvers in dangerous or restricted waters make it imperative to maintain a positive couple between the motors and generators. To make the relay inoperative, it is merely necessary for the operator to press a button, or close a small switch which bridges the overload relay contacts.

"N"—Three pole, single throw, knife switches for connecting the auxiliary generators to the main bus. Three pole switches are provided to permit parallel operation.

"O"—Overload time limit circuit breakers for all generators when connected to the auxiliary bus.

"P"—Two pole, transfer switch for excitation circuits. This switch is very similar to "A," except that both throws are like the lower throw of "A" and have the preventative resistance.

The purpose of the switch is to provide a ready means for quickly transferring the excitation circuits from one auxiliary generator to the other, and also to provide an excitation connection to the auxiliary generators which is unprotected by circuit disrupting devices.

Figure 7 shows a front view of a switchboard and control panel for a small Diesel electric drive, consisting of two generator units and one double unit motor operating on the principle herein described. It will be noted that the generator field control handle is mounted directly on this switchboard and is shown in the center near the top. The switches

for cutting in and out the various generators and motor units are shown on the center panel. In this particular case, however, the switches are not of the transfer type, and switching must be done on dead circuit.

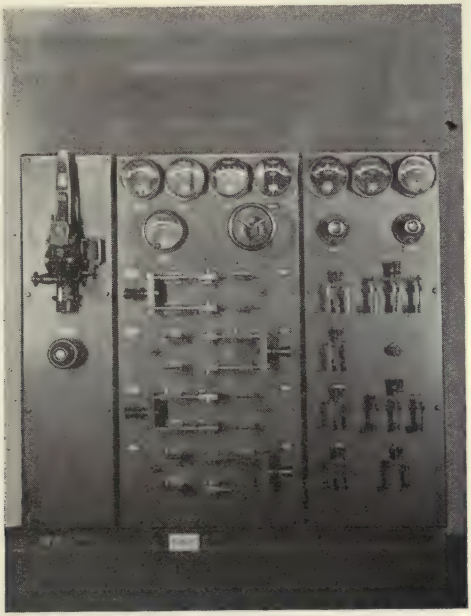


Fig. 7—Switchboard and Control for a Single Screw, Diesel Electric Drive

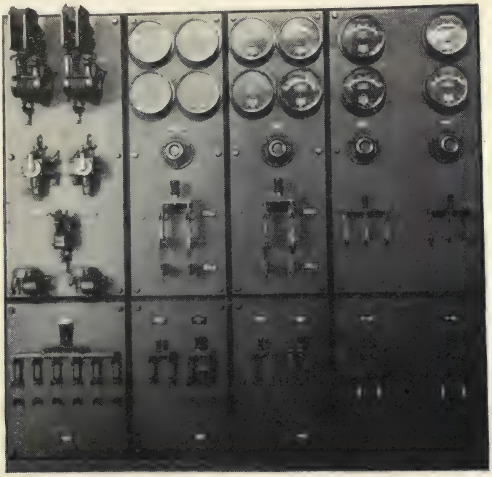


Fig. 8—Front View of Switchboard for a Double Ended Ferry Boat, Diesel Electric Drive

Figure 8 shows the front view of a switchboard for a double-ended ferry boat Diesel electric drive. The control pedestal for operating the field rheostat is located in the pilot house.

Figure 9—This view shows the rear of the board, the front view of which is shown in Figure 8.

Figure 10—This view shows a double face plate rheostat of the type used in the main generator circuit for controlling the propeller motor speeds. The rheostat is actuated manually from any remote point.

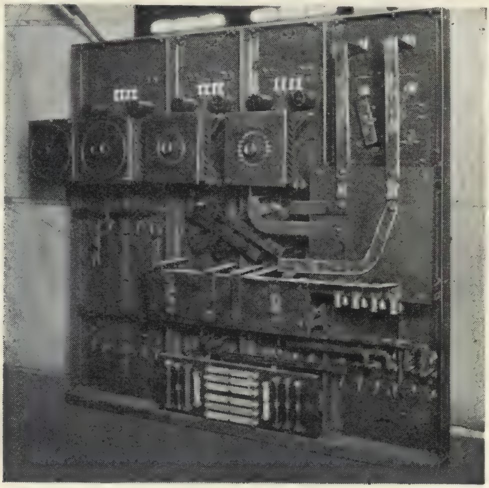


Fig. 9—Rear View of the Switchboard Shown in Fig 8

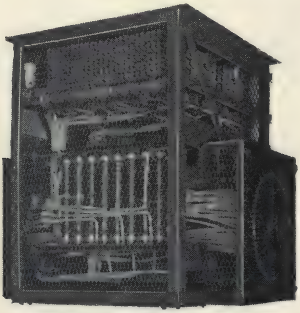


Fig. 10—Double Face Plate, Main Control Rheostat

Figure 11—This photograph shows a view of the generator field control pedestal for a drive having two motors independently controlled. All maneuvers of the ship are effected by motion of the small levers shown at the top of the pedestal.

Figure 12 shows a view of a control pedestal for a single screw ship such as is described.

Fig. 11—Control Pedestal for Twin Screw Diesel Electric Drive

Fig. 12—Control Pedestal for a Single Screw Diesel Electric Drive

Operation

Preparing to Get Under Way: Upon receipt of notice to prepare for getting under way at full power, the first operation is to see that all switches are in the proper position. Close switches "A" and "B" in upper position; close switch "D" to the right; close field circuit breakers "F"; close excitation switch "P" to the exciter which is in operation; see that generator field control handle is in the "off" position, and start the engines.

Getting Under Way Ahead: Upon signal to get under way, close the field switches "E" and "J". (The closing of "J" establishes power to the circuit breaker closing coil.) Move control handle ahead in answer to the signal and adjust the speed to the required value.

Getting Under Way Astern: Proceed as under way "ahead," as described above, except move the control handle to the astern direction.

Stopping: If it is merely desired to stop the ship without regard to time, move the control handle slowly to the "off" position.

If it is required to stop the ship quickly as in the case of an emergency, move the control handle toward the "off" position at such a rate as will maintain the current at 100 per cent to 150 per cent of full load

value. In this operation, the current will reverse as soon as the generator voltage is reduced below the motor counter voltage, and will stay reversed until astern operation ceases.

Special Set Ups: If the machinery is idle when changing set ups, the switches may be thrown without any special precautions. If, however, the ship is under way at the time of changing set ups, reduce the voltage by means of the individual field rheostat to approximately zero, and trip the field circuit breaker of the generator which is to be taken out of service. Then throw switch "A" to its extreme lower position after the electric lock described above has released the switch lever. Then shut down the engine.

To put a generator back into service while the ship is under way, reverse the above operations.

Since the probability of having to take a motor unit out of service while the ship is under way, is extremely remote, no provision is made for disconnecting it without interrupting the circuit. To take a motor unit out of circuit, move the main generator field control slowly to the "off" position; open the generator field rheostat switch "J" and the proper motor field switch "E"; then throw the proper switch "B" to the lower position. To re-establish the power, close the switch "J" and operate the control handle in the normal manner. When operating with one motor, it is unnecessary to use more than two of the generators.

Other set ups must be performed in a similar manner.

Securing Electrical Machinery While in Port: When the ship has docked, close all generator switches "A" in the full lower position, open motor switches "B," open generator field switches "D" and generator field breakers "F," open generator field rheostat switch "J" and open motor field switches "E."

If the machines are to be idle for more than 24 to 48 hours, and it is expected that they will be subject to appreciable changes in temperature, or to cool noticeable below the ambient air ,their fields should be excited at a low value to prevent sweating (condensation of miosture on the windings). A separate circuit (not shown in the diagram) is usually provided for this purpose. Although the insulation is made as moisture resistant as practice permits, the windings should not be allowed to sweat or become wet.

Port Operation:

(A) To use the main generators in port, for cargo winches and auxiliary power purposes, the following operations are necessary:

1. Start the engines which are to be used.

2. Close field switch "D" to the left for self-excitation.

3. Close circuit breaker "O."

4. Build up voltage to normal by cutting out resistance of field rheostat. (If the voltage builds up in the wrong direction, open switch "D," and close it to the right, close switch "J" and build voltage up in proper

direction to about half value, by moving the main control handle.) Open "J" and close "D" to the left, and repeat as above.

5. Close switch "C," thus connecting the generator on auxiliary bus. Adjust voltage.

(B) To parallel a second main generator:

1, 2, 3, 4. Same as above.

5. Be sure that the machine voltage is the same as the bus voltage, and then close switch "C."

6. Adjust voltage so that the machines divide the load as indicated on the ammeters.

(C) Subsequent machines are paralleled in the same manner as described under "B."

(D) If one of the auxiliary sets is in operation when the first main generator is connected to the auxiliary bus, it must be paralleled as in "B."

(E) The auxiliary sets are paralleled with each other and with the main generators on the auxiliary bus as described in "B."

CONTROL EQUIPMENT FOR CARGO SHIP WITH FOUR 500-KW 250-VOLT GENERATORS AND ONE 2,500-H.P. DOUBLE ARMATURE PROPELLING MOTOR

(General Electric System)

The equipment to control the operation of the electrical propelling machinery consists of the apparatus necessary to start, stop, reverse and vary the speed of the propelling motor and to cut out of circuit, any of the generators or either of the two motor armatures. Exciter panels are provided to enable the operator to adjust the voltage of the exciters and connect them so as to furnish excitation for the motor and generator or to furnish auxiliary power.

The following pieces of apparatus make up the control equipment:

One Controller

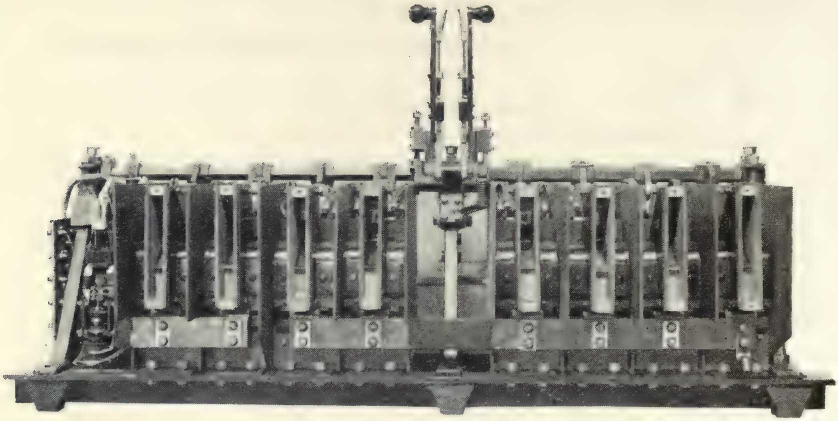
One Generator Field Rheostat

One Main Control Panel

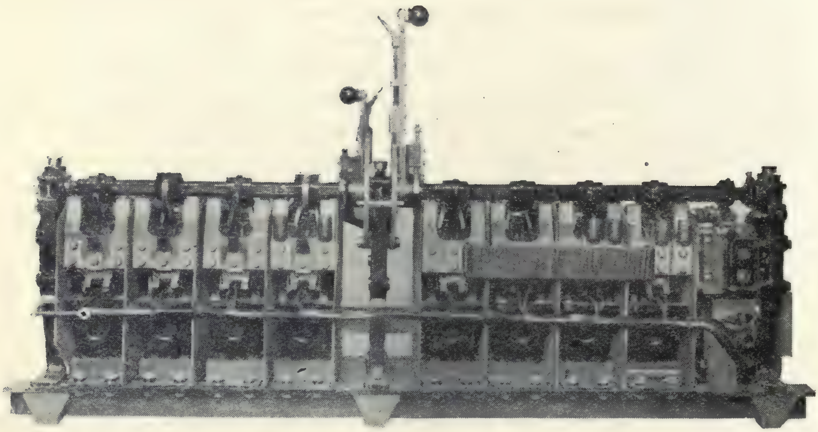
Two Exciter Panels

One Propeller Speed Magneto.

Controller: The controller consists of a cast iron frame with a sheet metal cover in which are mounted the main control drum and the motor field drum. The construction of the frame and cover is such as to make the controller practically water-tight; the cover clamping against felt in a groove in the frame. The control wiring is taken out of the controller at the bottom through the base. There are two operating levers, one for rotating the main control drum and the other for operating the



DJS-110 B Equipment of Control Group—Back End View. Demonstrating Neutral Position.



Control Group, Back View. (General Electric Diesel-Electric System) Showing Interlocking Levers for Hand Control.

motor field drum. The motor field lever has two positions, off and on, and is interlocked with the main lever so that the main lever cannot be operated unless the field lever is in the on position, and also so that the field lever cannot be turned off unless the main lever is in the off posicuit, sufficient resistance to limit the generator voltage.

The main lever when turned from the off position, closes the field circuit of the generators and as the lever is turned on the resistance in series with the generator fields is cut out of circuit. To reverse the motor, the main lever is turned off thru the off position to the astern points. In passing through the off position the generator fields are reversed and the generator voltage is increased as the resistance is cut out of circuit.

Generator Field Rheostat: The generator field rheostat is composed of ribbon wound on resistor units mounted in boxes with taps brought out which are connected to the fingers of the controller. The boxes in which the units are mounted are made up of perforated sheet metal with an insulated base on which the terminals are mounted.

Control Panel: On the control panel are mounted instruments necessary to indicate the generated voltage and current and the propeller speed. Also the switches required to connect the generators and motor armatures in series and to cut out of circuit, any of the generators or either motor armature. In addition to the apparatus just listed, field contactors for the motor fields are supplied with their discharge resistances, an overload relay and generator field contactor with resistance so arranged that in case of over-load the relay will trip and cut into circuit, sufficient resistance to limit the greater voltage.

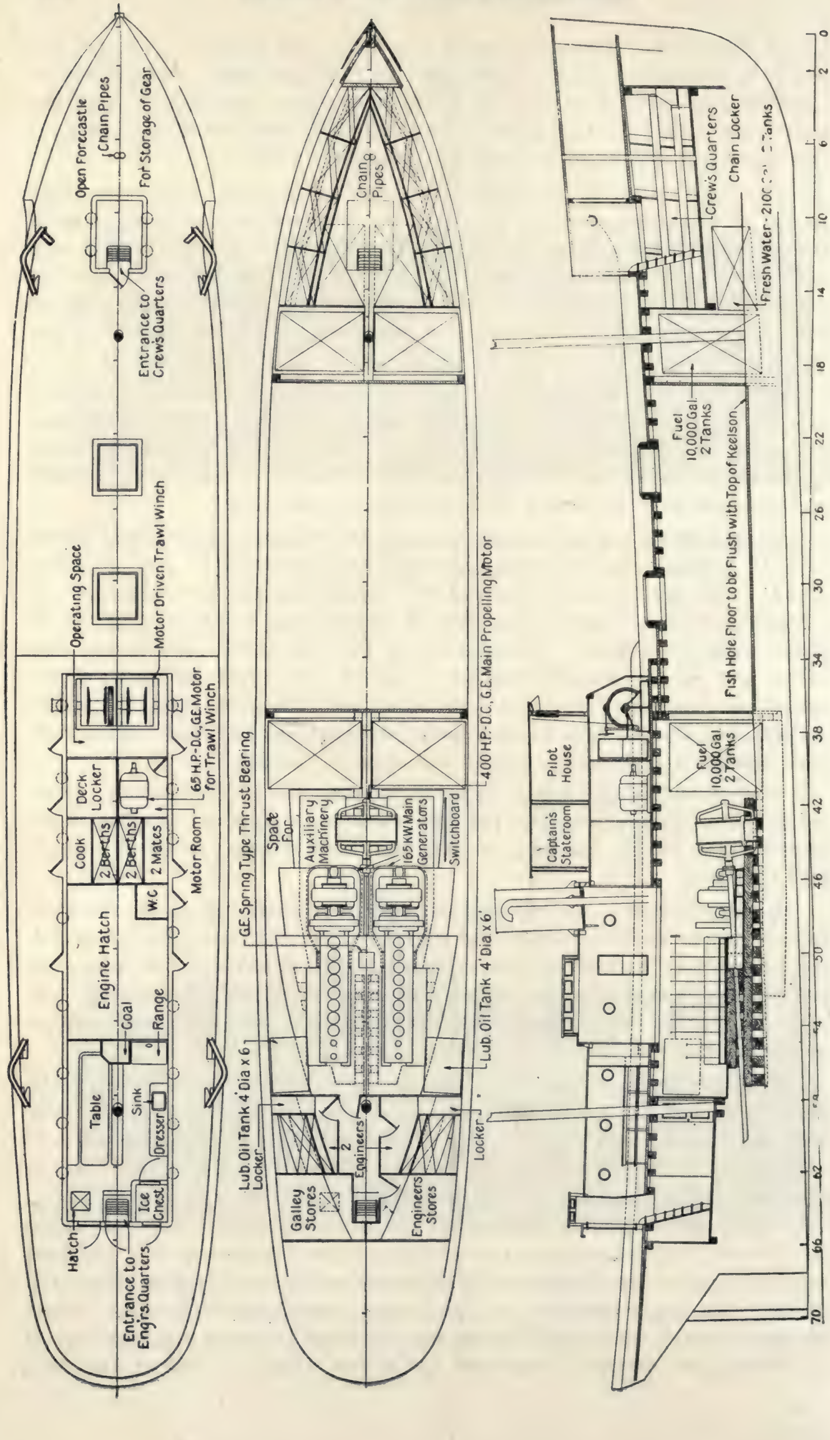
Connected to the panel, there is mounted a line contactor so arranged that during reversal the line current is limited to a pre determined value by the opening of this contactor which inserts resistance into the line.

Exciter Panels: On the exciter panels are mounted instruments for indicating the exciter voltage and current; the field rheostats for adjusting the voltage as well as the overload circuit breakers with necessary switches for connecting exciters either to the bus which provides current for exciting the generator and motor fields or the bus which supplies power for the auxiliaries.

Propeller Speed Magneto: The propeller speed is indicated by the speed indicator on the control panel which is connected to a small magneto driven by gears from the propeller shaft. The instrument is calibrated so as to read directly in R.P.M., since the voltage delivered by the magneto varies directly as the speed.

Operation of Control Equipment: With the apparatus correctly wired up and the engines running the voltage on the exciters is first adjusted and the switch closed so as to provide excitation for the generators and motor. The main switches on the control panel should then be closed so as to connect generators which are to be run in series with the motor.

When the signal is received for either ahead or astern the field



General Plan View, Illustrating Arrangement of Propelling Machinery.

lever on the controller must first be placed in the on position. This operation energizes the motor field contactors which close and energize the motor fields. With the field lever "on" the main lever can be moved either ahead or astern and the voltage of the generators adjusted so as to give the desired motor speed. For high speeds, the rated motor current should not be exceeded. The operator should move the main lever to that point which gives full load current.

In case of overload, the generator voltage is reduced through the action of the overload relay which causes the generator field contactor on the control panel to open and insert a limiting resistance in the generator field circuit. When the overload is removed the relay is returned to its operating position and the contactor closes, cutting out the resistance.

In reversing, when the main lever is returned to the first point the line contactor is opened and a limiting resistance inserted into the line. This resistance is in circuit when the main handle is in the first position ahead, off position, and first position astern.

DIESEL ENGINE—ELECTRIC DRIVE

The "Mariner"—The First Electrically Operated Trawler.

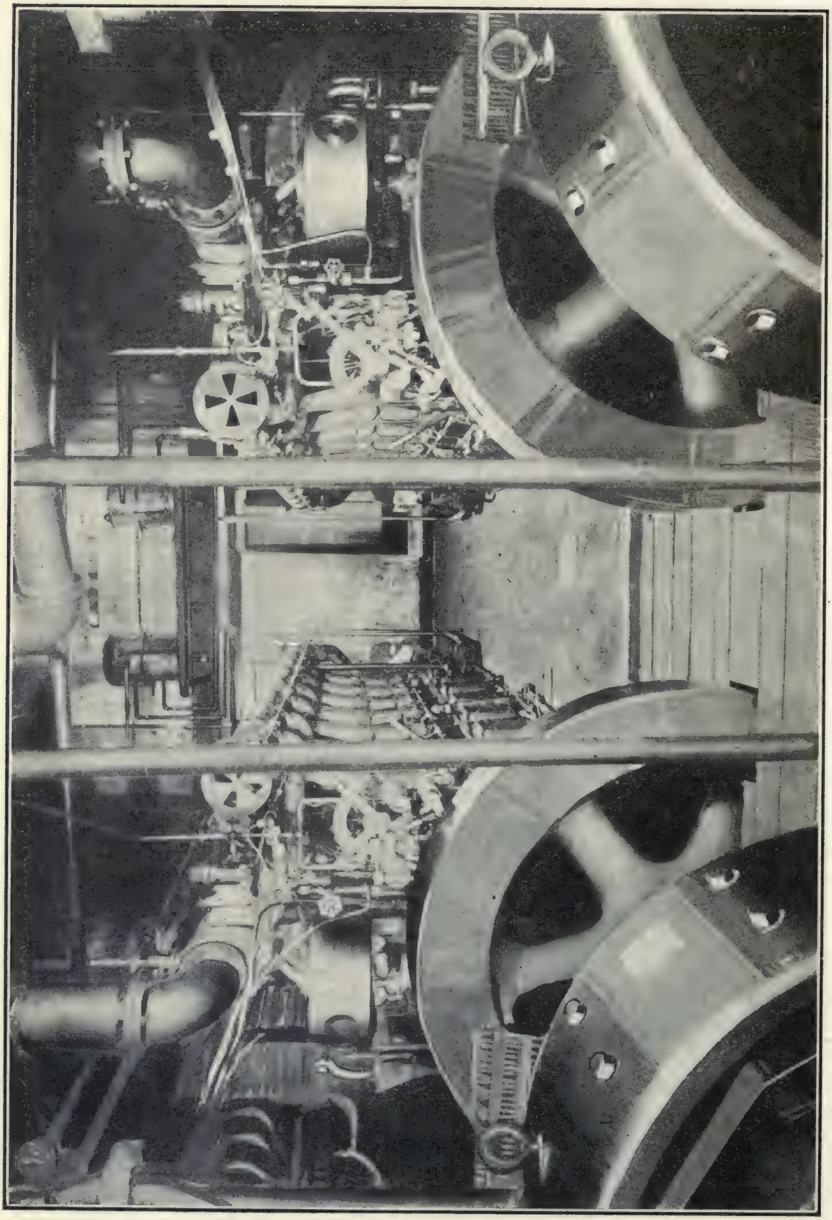
The adoption of electric propulsion for the beam trawler "Mariner" was the logical result of the efficient and economical operation secured with this system in numerous crafts of various kinds, both in America and Europe during the past twelve years.

In designing the equipment for the "Mariner" the inherent flexibility of the electrical method of power application made it possible to obtain high economy in fuel consumption, especially under cruising conditions, sustained uniform rate of rotation for the engines, positive control of the propeller speed at all times, a high factor of safety by means of three separate control stations, practically instantaneous reversal of the propeller, and the use of electric motors for driving auxiliaries such as pumps, compressors, hoists and ventilating blowers.

Hull: The craft is of wooden construction, and is rated at 500 tons, with dimensions as follows: Length over all, 150 ft.; beam, 24 ft. 3 in.; mean draft, 11 ft. 9 in. Her cruising radius at 10 knots is 6,000 miles, and at three-quarter speed 9,000 miles.

Propelling Equipment: The propelling equipment comprises two eight cylinder, four-cycle, 350 R.P.M. Diesel engines, each direct-connected to a 165-kw., 125-volt, direct-current generator. The two self-excited generators are normally connected in series and supply current to a 400 H.P., 250-volt, 200 R.P.M. motor, which is direct-coupled to the propeller shaft.

Two control stations are located in the engine room—one provided with remote control, and one arranged for emergency manual operation; a remote-control outfit is also located in the pilot house.



Engine Room (looking forward), showing arrangement of Diesel Engine allowing excellent view of the Main Generator. Note Main Generators with the Engine Fly Wheels carried on the Generator Shaft Bearings.

Both the generators and the motor are designed specifically for sea duty, and are provided with non-corrodible fittings and heat-resisting insulation throughout. The bearings are a combination of waste-packed and oil-ring type, with special provision against the leakage of oil along the shafts, when the machines are out of their normal positions, due to the rolling and pitching of the ship. Finally, armatures and fields are water-proofed, and the machines are so designed as to prevent flashing in the presence of moisture, due to either atmospherical conditions or flooding of the engine room in rough seas.

In order to insure ample mechanical strength for the electrical machinery, steel castings were used for all rotating parts which would be subjected to unusual strains, or to shocks incident to operation during stormy weather.

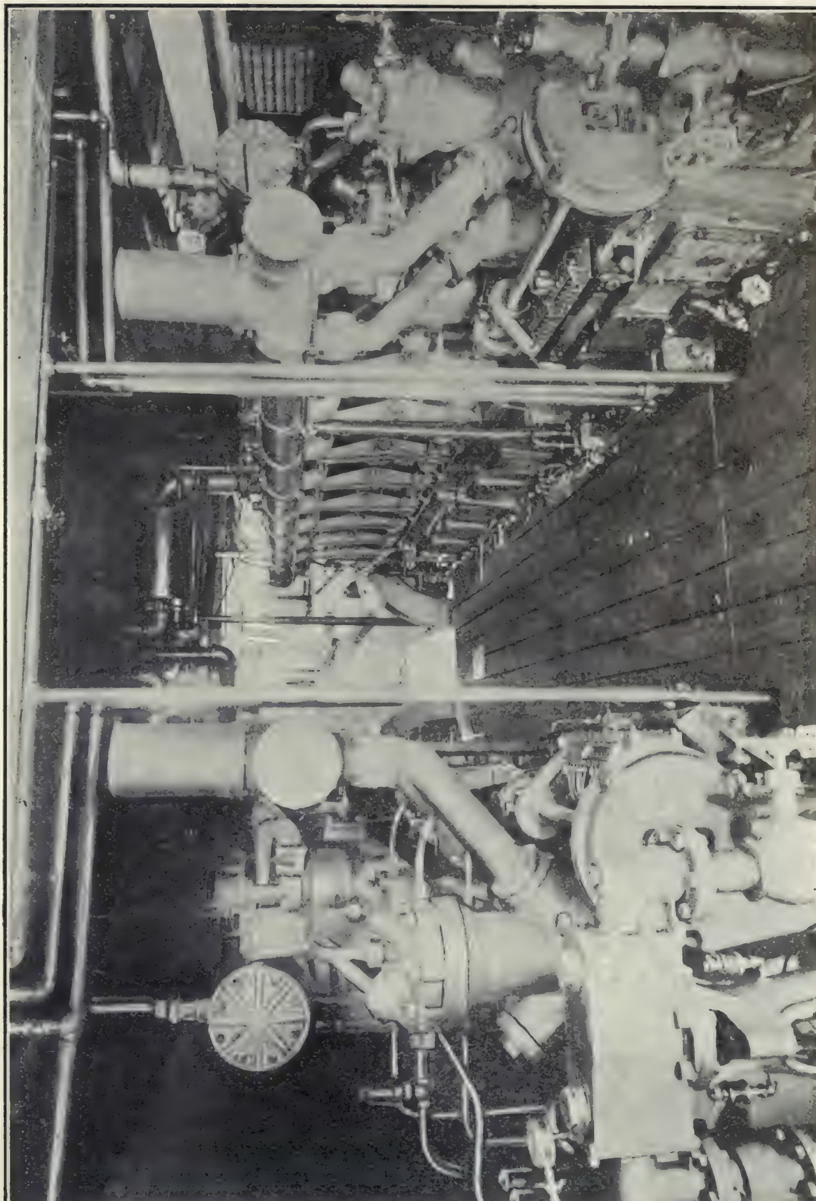
The 400 H.P. propeller motor is located forward of the generating sets and has a normal full-load speed range of from 160 to 200 R.P.M. It is a compound wound machine and, when taking current from both generators, it operates at 250 volts; but, for slow cruising, one engine can be shut down and the motor then receives current at 125 volts. Under these conditions it has a speed range of from 70 to 160 R.P.M.

Propeller: The propeller is 94 in. in diameter by 68 in. pitch and, at full-load rotation of 200 R.P.M., gives a speed of between 7 and 10½ knots, depending upon weather conditions. When hauling the net the full horsepower of the motor is developed at a propeller speed of 160 R.P.M.

Control Equipment: Engine room control of all electrical circuits is secured by means of a main panel board, on which are mounted the engine-room meters, generator field switches and resistors, switches and fuses for the propelling and auxiliary motors, and an overload relay for the main hoist motor. The meters are mounted at the top of the panel and are special instruments designed for shipboard work, being equipped with moisture-proof, non-corrodible parts. The dials are black with white markings, with radium paint on the needles and dial markings. A duplicate set of these instruments is installed in the pilot house.

The starting resistor resistance consists of five boxes of grids, which are mounted on the starboard side of the engine room. Just forwards of these grids, the control contactors are located. This group consists of the necessary curren-carrying contactors for starting, stopping and reversing the motor, an overload relay and motor-shunt field discharge resistance, and is normally operated by means of one of two master controllers—one located in the engine room and the other in the pilot house.

During operation from either of these master controllers, the contactors are closed magnetically; but, if for any reason they cannot be operated magnetically, handles attached to camshafts are provided which may be operated manually to close the contactors in the desired sequence. The overload relay, in case of overload, opens the circuit through the reversing contactor coils, causing them to open the line circuit. The handles for manual operation are so interlocked that the reversing handle must be operated before the accelerating handle; therefore, the acceler-



Engine Room (looking aft), showing arrangement of Diesel Engines Driving the Main Generators.

ating handle must be turned off before the reversing handle can be moved. This arrangement insures absolute safety for the control system of the ship, even in the very improbable event of failure of, or injury to, the two remote control equipments.

One of the important advantages of electric propulsion is that of remote control, which permits the actual maneuvering of the ship, to be accomplished directly in the pilot house, if desired, without the necessity for signals to the engine room. At sea, this remote control system is normally only a convenience, as compared with the ordinary combination pilot house and engine room control; but, in entering and leaving slips, in congested harbors, in narrow and swift current waterways, and for quick reversal or change of speed in emergencies, its great practical value is obvious.

This type of master controller installed on the "Mariner" consists of a cast-iron frame, with a sheet-metal cover, in which are mounted a main control cylinder and a reversing cylinder. The construction of the frame and cover is such as to make the controller practically water-tight the cover lamping against felt in a groove in the frame. The control wiring is taken out of the controller at the bottom through the base.

There are two handles on the controller—one main and one reversing. The main handle rotates the main cylinder, which gives 17 operating positions—one off position and one overload relay reset position. The reversing handle rotates the reversing cylinder and has three positions; ahead, off and astern.

These two handles are so interlocked that the main handle cannot be moved beyond the overload relay reset position unless the reversing handle is in either the ahead or astern position, and so that the reversing handle cannot be moved unless the main handle is in either the off or reset position.

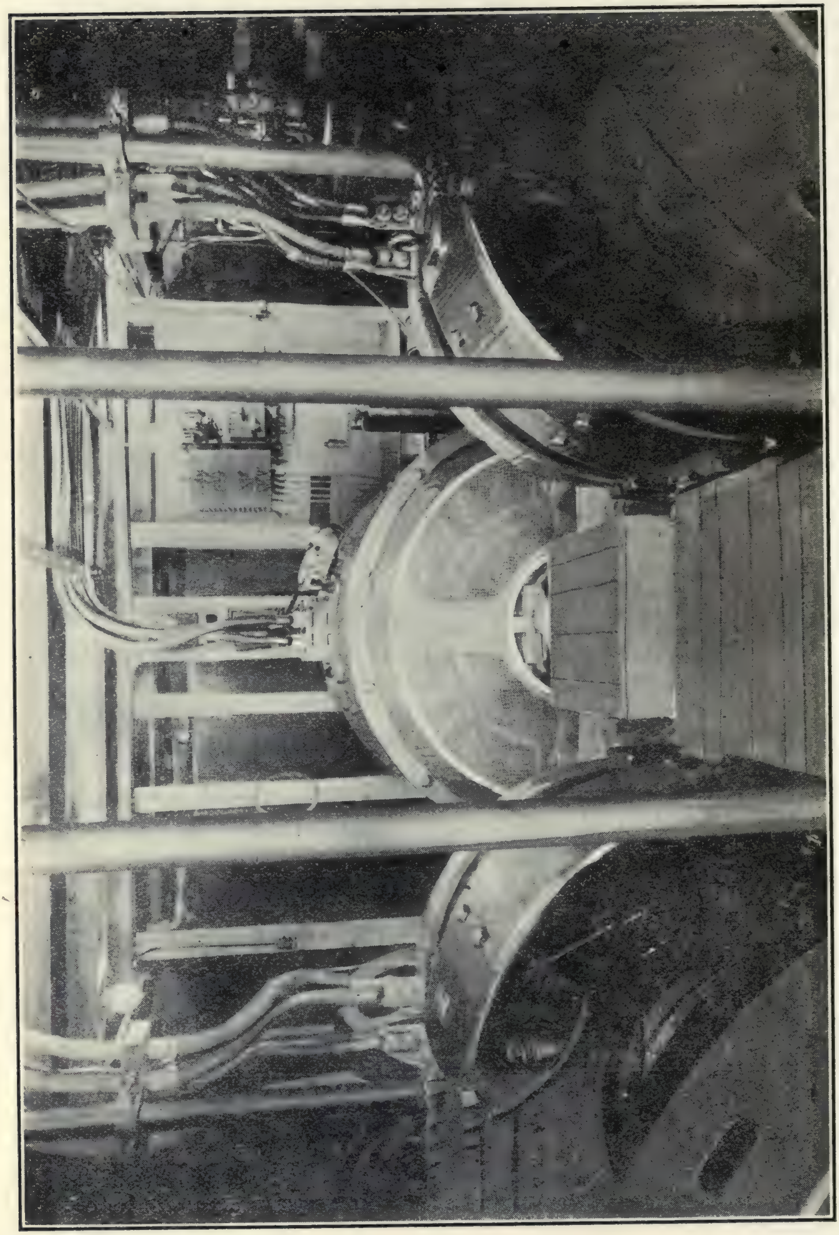
The rapidity with which the motor-driven propeller can be reversed was demonstrated during the first trial trip when, with the propeller rotating at from 193 to 196 revolutions per minute, it was reversed from full speed ahead to full speed astern in 13 seconds; the actual reversal of current in the motor being accomplished in two seconds.

Pilot-house control was used throughout the test run, operating either one or both generators with equal facility.

Ship's Thrust Bearing: Instead of the usual rigid multi-collar type of thrust-bearing, a self-oiling spring thrust bearing of the single collar, self-aligning type is used, located aft the driving motor and sustaining a thrust of 7,500 lbs., with the propeller revolving at 200 R.P.M.

Electric Auxiliaries: In addition to operating the propelling equipment, electrical energy is used for lighting and all auxiliary power purposes and, when the main engines are shut down, current is supplied by means of an independent 15-kw., 125-volt, oil engine driven generator, installed in the forward end of the main engine room on the port side.

The emergency air compressor outfit is driven by a direct-geared motor and is provided as an insurance against the improbable loss of



Forward End of Engine Room of Trawler "Mariner" (looking forward), showing Main Generator and Propeller Motor with Master Controller at Right.

starting air for the engines. Under these conditions it will be utilized to fill the air-starting bottles, as the auxiliary generating set can be started by hand.

The bilge and water-supply pumps are small centrifugal units, each driven by a direct-coupled motor, and near the main generators and propeller motor a small motor-driven ventilating set is utilized to prevent excessively high temperature in the engine room.

The fishing operations are carried on by means of a 65 H.P., motor driven, main double-drum hoist, installed on the main deck forward of the engine room, which handles the haulage cables and ropes of the net as they pass through the hoist brackets fore and aft on either side. The unloading of the fish at the dock is accomplished by means of a 5 H.P. motor-driven whip located near the forward mast.

The boat has proven a success and marks a new epoch in motor-driven ships. She was built by Arthur D. Story, of Essex, Mass. The engines were built and installed by the New London Ship & Engine Co., of Groton, Connecticut. All other equipment was supplied by the General Electric Company of Schenectady, N. Y.

CONTROL EQUIPMENT FOR THE ELECTRICALLY PROPELLED M. S. "FORDONIAN"

(General Electric System)

The main control equipment, provided to control the operation of two Diesel engine-driven generators and the propulsion motors consists of a control panel on which are installed the various switches, field rheostats, and instruments; a master controller for operating the contactors of the control group in the proper sequence; a control group for starting and reversing the motors; a starting resistor in connection with the control group used when starting the motors; and a resistor in the fields of the motors used to obtain speeds of the motors of about 60 to 90 R.P.M. when operating with only one generator.

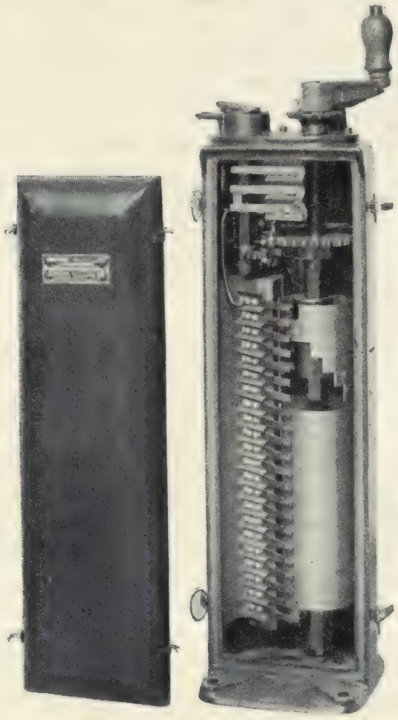
The switches and wiring are so arranged that either one or both generators can be used to supply line current to the propulsion motors. Constant normal voltage is held on the generator, which is chosen to supply excitation current to the fields of the generators and motors, and current to the blower motor, auxiliary panel and control circuit, while the voltage of the other generator can be varied to obtain different speeds of the propulsion motor. By this flexibility of operation, emergency conditions can readily be taken care of.

This control equipment is installed in the engine room and consists of the following:

- One Control Panel
- One Master Controller
- One Control Group
- One Starting Resistor
- One Motor Field Resistor

Master Controller: The master controller consists of a cast iron frame in which are mounted a main control cylinder and a reversing cylinder. A sheet metal cover is clamped securely against felt in a groove in the side of the frame so as to make the controller splash-proof. The control wiring is taken out through the base of the controller.

There are (2) handles on the controller, one main and one reversing handle. The main handle rotates the main cylinder through an arc of 330 degrees, thereby covering an "off" position, an overload relay "reset" position and (5) contactor positions and (12) motor field resistor posi-



Master-Controller, Type C-S 143 A, Specially Designed for Marine Use. Controller Used in Pilot House and Engine Room.

tions. The reversing handle rotates the reversing cylinder and has three (3) positions: "ahead," "off" and "astern."

These (2) handles are so interlocked that the main control handle cannot be moved beyond the overload relay reset position unless the reversing handle is in either the "ahead" or the "astern" position, and so also that the reversing lever cannot be moved unless the main handle is in either the "off" or "reset" position.

In ordinary operation, the main handle will be thrown only to the "reset" position and not to the "off" position except when the motor is

to be stopped for some time or the switch positions are to be changed. To throw from the "reset" to the "off" position, a latch in the handle must be released. This latch is provided to prevent going into the "off" position, and thus opening the motor shunt field every time the motor is stopped or reversed, since the motor shunt field contactors are always closed except when the controller handle is in the "off" position.

Control Group: The control group is designed for mounting from above, and contains four (4) main contactors for completing the motor line, two being used for ahead and two for astern operations; four (4) contactors to reduce and cut the starting resistance out of the circuit; one (1) overload relay to trip out the main contactors when the load is excessive; two (2) motor field contactors and the motor shunt field discharge resistance.

The main line and starting resistor contactors are designed to carry the full load current of 1400 amperes and during normal operation are closed magnetically by means of solenoids energized from circuits through operation of the master controller. If for any reason the contactors cannot be operated magnetically, handles are attached to cam shafts by means of which the contactors can be closed manually in the proper sequence.

The overload relay is composed of one series coil with armature and one shunt coil with a plunger carrying two disk interlocks. When the relay is in the tripped position the lower disk closes a circuit through the shunt coil of the relay which circuit is completed through the master controller in the reset position. The completion of this circuit picks up the plunger of the relay so that it is latched up, thereby closing the other disk interlock which is in the solenoid circuit of the four line contactors. This solenoid circuit is completed in the controller in all positions of the main cylinder except the "off" and "reset" positions.

In case of overload the excessive current through the series coil picks up the armature and thereby releases the plunger carrying the interlocks, thus causing the solenoid circuit to be opened and opening the two line contactors which are in use.

When the contactors are operated manually they are held closed by cams. Interruption of the solenoid circuit by tripping of the overload relay therefore does not cause the contactors to open, and for this reason extreme care must be used when the contactors are closed by hand so that the load on the motors will not exceed the normal current value.

The two (2) field contactors close the two motor shunt field circuits magnetically on the "reset" point of the master controller and close the field circuits mechanically when operated manually.

The discharge resistance is connected permanently across the shunt fields of the propelling motors to limit the inductive kick when the field contactors open.

The two (2) handles for manual operation of the control group are so interlocked that the reversing handle must be operated before the

accelerating handle and so that the accelerating handle must be turned off before the reversing handle can be moved.

Control Panel: On this control panel are mounted the engine room instruments; the generator cut-out switches, field switches, and rheostats; the combined line and field switches for the main propelling motors; the switches for the blower motor, auxiliaries and the excitation supply to the controller. There are also mounted the necessary shunts and generator field discharge resistors.

The instruments are mounted at the top of the panel and are designed for marine use, being equipped with non-corrosive parts. The dials are white with black markings. Each generator line is equipped with a voltmeter having a 0 to 300 scale, and also with a 0 to 2500 scale ammeter mounted. A millivoltmeter with a 150-0-150 R.P.M. scale for propeller speed indication either ahead or astern and an ammeter with a 0 to 100 scale for measuring the field current of the motors are mounted directly beneath the generator ammeters.

The two generator field rheostats, made up of form R, size U wire and ribbon wound resistance units are supported by the panel frame above the instruments. They are operated by the rheostat handles through means of chains and sprockets. The total resistance of each rheostat is approximately 50 ohms, which when inserted reduces the generator field current to about 4 amperes.

The generator field rheostat handles and field switches are mounted below the instruments so that the operator in adjusting the generator fields can readily watch the meters.

There are two single pole, double throw 1600 ampere switches to connect either or both generators across the propelling motors. Between these two switches is located a double pole, singel throw switch and fuses for the blower motor.

At the lower portion of the panel and near the center line are mounted two (2) double pole, double throw switches with fuses, in each throw. The switch to the right, facing the panel, is used to transfer the field excitation of the generators and motors and the blower motor and control current from one generator to the other. The switch to the left transfers the current supply to the auxiliary panel from one generator to the other. These double pole, double throw switches should not be opened until the generator field switches have been opened, and in the case of the switch supplying the auxiliary panel all switching on this auxiliary panel should be properly made before transferring the circuit.

On either side of the above switches are mounted two double pole, double throw switches for cutting out either of the motor armatures and fields in case of an emergency. One side of the switch is in the motor line while the other is in the motor field so that it is impossible to close the motor line switch without also closing the motor field switch. These switches should not be opened unless the main handle of the controller and the reverse handle of the control group are both in the "off" position, accomplished by moving the master controller handle to the

"off" position. Suitable barriers are placed between the motor switches and the excitation and auxiliary switches. Straps are placed on the back of the panel near the generator and motor cut-out switches and motor field switches and in their respective circuits to be disconnected as explained later for emergency operation.

No work must be done on the generators or motors except when the machines are thoroughly isolated by disconnecting all leads at the terminals of the machine in question. The leads should be properly taped and tied to prevent them from swinging about.

Starting Resistor: The resistor used in starting the motors is made up in four (4) sections connected, and is composed of five (5) boxes of IG grids.

The first section of the resistance has a resistance of .36 ohms and is made up of 36 No. 61 IG grids in series.

The second section is cut in on the 2nd point of the master controller, in parallel with the first, and has a resistance of .18 ohms. It is composed of 18 No. 61 IG grids connected in series. The total resulting resistance of the resistor on the second point is .12 ohms.

The third section of resistance is made up of 24 No. 61 IG grids, two in parallel, and has a resistance of .06 ohms. This section is cut in on the 3rd resistance point of the controller and is in parallel with the first two sections, giving a resulting resistance of the resistor of .04 ohms.

The fourth section is composed of 12 No. 62 IG grids connected three in parallel, and has a resistance of .02 ohms and is cut in on the 4th point in parallel with the other three sections, the resulting resistance being .0133 ohms.

On the next or 5th point of the controller, all the resistance is short-circuited.

Each of the 5 boxes contains 18 grids, and is made up in a frame. The box is designed for mounting with the grids carefully spaced in a vertical position, thereby giving the best ventilation.

When operating with only one generator supplying current to the motors, it is possible, though not advisable, to operate continuously on any step of the rheostat, but when both generators are being used in series with the motors, the length of time which the controller handle can be safely held on any of the first four points is as follows:

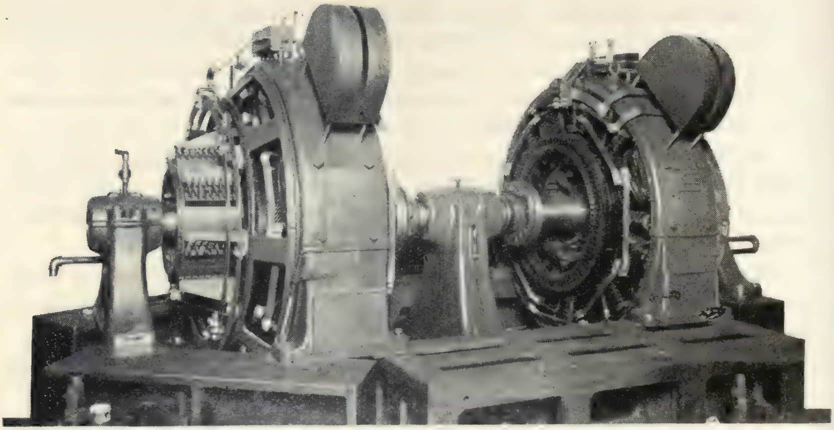
Point No. 1—20 seconds should not be exceeded.

Point No. 2—45 seconds should not be exceeded.

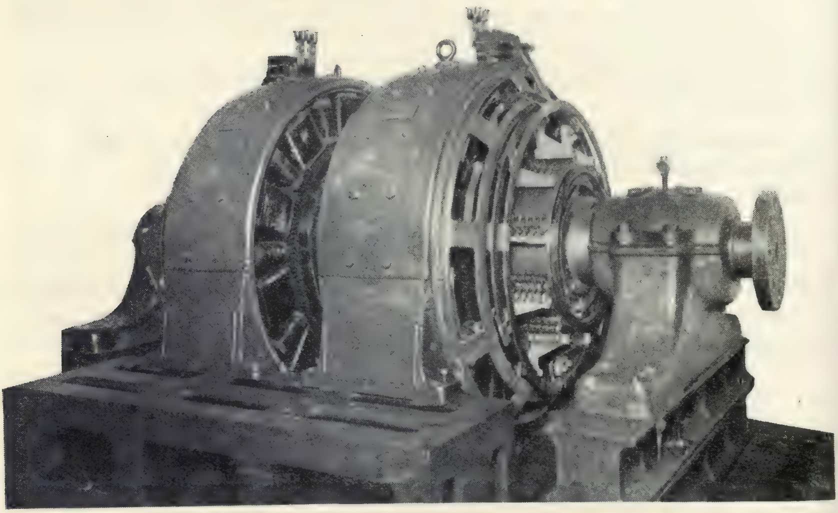
Point No. 3—60 seconds should not be exceeded.

Point No. 4—2 minutes should not be exceeded.

The above limitations are derived from data available before installation. Actual operating conditions might permit an extension of the limits given above, but if these limits are exceeded, the grids of the starting resistor should be carefully watched for overheating.



Two Marine Direct-Current Generators (for M/S. "Fordonian") Rated MPC-8 Pole 350 KW 200 R.P.M. Volts Compound-Wound on the Testing Stand.



Marine Direct Current Double Armature Motor (For M/S "Fordonian") Rated 850 H.P., 120 R.P.M., 500 Volts, Consisting of Two MPC-10 Pole 425 H.P., 120 R.P.M., 250 Volt Shunt Wound Motors mounted on one shaft. Port Looking Forward.

Resistor for the Felds of the Motors: The resistor for the two motor fields in parallel is composed of 8 units mounted in a box frame, with 13 taps brought out, that is, there are 12 sections of resistance. This resistance is required when the motor is running on 250 volts to increase the speed to approximately 90 R.P.M.

The resistance of the 8 units in the order in which they are cut into the circuit, is as follows: 0.52 ohms; 0.52 ohms; 0.52 ohms; 0.85 ohms; 0.85 ohms; 1.7 ohms; 1.7 ohms; 1.7 ohms; making a total resistance of 8.36 ohms. The resistor is designed to vary the field current for the two fields in parallel from 35 amperes at full field, to 17 amperes at the 90 R.P.M. speed on 250 volts.

The terminals of this resistor are connected by a cable to the master controller terminals. The resistance is cut into the field circuits as the main handle of the controller is advanced beyond the fifth point.

Operation of the Control Equipment: In preparing to get under way with the apparatus wired up, either or both Diesel engines are brought up to speed, depending on whether the propelling motor is to be run on 250 or 500 volts. If both generators or 500 volts are to be used, the excitation switch and both generator field switches should be closed and the voltage adjusted with the generator field rheostats to give 250 volts on each generator. With the excitation switch thrown down, generator No. 1 supplies current for the fields of the generators and motors and the blower and control circuit, but when in the "up" position, generator No. 2 is the source of supply. Both generator line switches and both motor line and field switches should next be thrown to the "up" position. This places the generators and motors in series. When the generators are thus connected to operate in series, the series field switches located on the generators should be opened if it is desired to obtain the full propeller speed of 120 R.P.M.

Start the blower motor for ventilating the propulsion motors and keep it running as lng as there is any current on the propelling motors. **Even though the motors might temporarily be at rest a condition exists when the controller handle is on the "reset" position, where full field is still maintained on the motors and it is very essential that ventilation be continued under this condition as well as when the motors are running.**

The motors are now ready for starting by means of the master controller. The reversing lever is set ahead or astern as desired and the main handle moved from the "off" to the "reset" position. The operator should notice the reading on the motor field ammeter to see that he has full field of about 35 amperes, which is the total current of both motor fields.

With this condition fulfilled, the main handle of the controller is then moved successively to points 1, 2, 3, 4 and 5, hesitating on each point for only a very short time, as explained under the heading, **Starting Resistor.** On point 1, if the ahead direction has been chosen, contactors 1 and 3 in the control group are closed. If the astern direction were chosen, contactors 2 and 4 would close. In either case the motors are connected

OPERATION CHART - NO. 1

NORMAL OPERATING CONDITIONS	PROPELLER SPEED - APPROX. R.P.M.	GENERATOR NO. 1 SWITCHES CUT-OUT	GENERATOR NO. 1 SWITCHES SHUNT FIELD	GENERATOR NO. 1 SWITCHES SERIES FIELD	GENERATOR NO. 1 RES RHEOSTAT	GENERATOR NO. 1 INS'MT VOLTMETER	GENERATOR NO. 1 INS'MT AMMETER	GENERATOR EXCITATION SWITCH	GENERATOR AUXILIARY SWITCH	GENERATOR NO 2. SWITCHES CUT-OUT	GENERATOR NO 2. SWITCHES SHUNT FIELD	GENERATOR NO 2. SWITCHES SERIES FIELD	GENERATOR NO 2. RES RHEOSTAT	GENERATOR NO 2. INS'MT VOLTMETER	GENERATOR NO 2. INS'MT AMMETER	BLOWER MOTOR SWITCH	MOTOR NO 1 SWITCHES CUT-OUT	MOTOR NO 2 SWITCHES CUT-OUT	MOTOR IN'ST AMPERES FIELD	CONTROLLER POSITIONS REVERSE	CONTROLLER POSITIONS OFF	CONTROLLER POSITIONS RESET	CONTROLLER STARTING AND RESISTOR POINTS STARTING POINTS 1 to 5	CONTROLLER STARTING AND RESISTOR POINTS RESISTOR POINTS 6 to 17	OVERLOAD RELAY	MS FORDONIAN DIESEL-ELECTRIC
OTH GENS AND OTH MOTORS FULL FIELD	105 to	2	5		7	7	14	4		3	6		8/15	8/15	14	9	10	11	13	12	1	13	14		13	SEQUENCE OF OPERATION
EN NO 1 AS EXCITER	120	U	C	O	Aj	250 volts	Rd	D		U	C	O	Aj	Aj for Speed	Rd	C	U	U	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
OTH GENS AND OTH MOTORS - FULL FIELD	105 to	2	6		8/15	8/15	14	4		3	5		7	7	14	9	10	11	13	12	1	13	14		13	SEQUENCE OF OPERATION
EN NO 2 AS EXCITER	120	U	C	O	Aj	Aj for Speed	Rd	U		U	C	O	Aj	250 volts	Rd	C	U	U	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
OTH GENS AND OTH MOTORS: FULL FIELD	90 to	3	6		8	8	15	5		4	7	2	9/16	9/16	15	10	11	12	14	13	1	14	15		14	SEQUENCE OF OPERATION
EN NO. 1 AS EXCITER EN NO. 2 SERIES FLD CUT OUT	105	U	C	O	Aj	250 volts	Rd	D		U	C	C	Aj	Aj for Speed	Rd	C	U	U	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
OTH GENS. AND OTH MOTORS-FULL FIELD	90 to	3	7	2	9/16	9/16	15	5		4	6		8	8	15	10	11	12	14	13	1	14	15		14	SEQUENCE OF OPERATION
EN NO 2 AS EXCITER EN NO 1 SERIES FLD CUT OUT	105	U	C	C	Aj	Aj for Speed	Rd	U		U	C	O	Aj	250 volts	Rd	C	U	U	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
GEN NO 1 AND	60 to	2	6	7	8	8	14	4		3	5					9	10	11	13	12	1	13	14	15	13	SEQUENCE OF OPERATION
BOTH MOTORS-WEAK FLD	90	U	C	O	Aj	250 volts	Rd	D		D	O	O				C	U	U	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
GEN NO 2 AND	60 to	3	5					4		2	6	7	8	8	14	9	10	11	13	12	1	13	14	15	13	SEQUENCE OF OPERATION
BOTH MOTORS-WEAK FLD	90	D	O	O				U		U	C	O	Aj	250 volts	Rd	C	U	U	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS

NOTE:— U = SWITCH CLOSED IN THE "UP" POSITION C = SWITCH CLOSED

D = SWITCH CLOSED IN THE "DOWN" POSITION O = SWITCH OPEN

AH'D = AHEAD AS'N = ASTERN Aj = ADJUST FOR VOLTAGE

Rd. = READ INSTRUMENT WHEN MOVING CONTROLLER HANDLE

across the two generators in series with the first step of resistance. Points 2, 3 and 4 of the controller close contactors 8, 7 and 6 in the order named, cutting in sections 2, 3 and 4 of the starting resistance in parallel with the first section. The 5th step of the master controller closes contactor 5, shorting out the whole starting resistor. This places the propelling motors across the two generators in series and under these conditions and with the ship moving at full speed, the motors will run at normal speed of 120 R.P.M., and will draw approximately full load current from the generators, depending upon the draft and other conditions. As explained later, the exact motor speed or current when running on both generators, is adjusted by changing the voltage of one of the generators.

The full load current of the motors and generators is approximately 1400 amperes, and this value should never be exceeded during continuous running conditions. Turning the controller handle beyond the 5th point weakens the fields of the motors and with both generators running at 250 volts the motors will probably draw considerably over full load current. For this reason, the operator should always watch the generator ammeters to see that the full load current is never exceeded. It will rarely be necessary to cut in any of the motor field resistance when both generators are supplying current, except when the ship is light loaded and more than normal speed is desired, in which case the controller handle is advanced over the motor field resistor points to obtain normal current and increased speed.

When operating under normal speed conditions, i. e., each generator supplying current at 250 volts, the load on each generator should be equalized as nearly as possible by taking excitation current off one generator and current for the auxiliary panel off the other generator.

To Operate With One Generator: If only one generator is to be used, the other generator is cut out of the circuit by throwing its corresponding single pole, double throw switch down to the "cut-out" position. The doublt pole, double throw excitation switch is then thrown down, if generator No. 1 has been selected, or up, if generator No. 2 has been chosen for excitation purposes. The field switch of the generator selected is then closed and the field rheostat adjusted to give 250 volts. The switches of the motors being closed in the "up" position, the master controller is set for ahead or astern and the main controller handle moved successively through the reset and first 5 points, thus placing the motors directly in series with the generator. The motors under these conditions run with full field at approximately 60 R.P.M. To further increase the speed of the motors the controller handle is moved over the remaining points, thus weakening the fields of the motors until full load current of approximately 1400 amperes is drawn by the motors, whose speed will then be about 90 R.P.M.

To Reverse the Motors: To reverse the motors the main handle of the controller must be thrown to the reset position. The reverse lever may then be thrown, after which the main handle may be advanced. This will bring the motors up to speed in the reverse direction.

OPERATION CHART – NO. 2

EMERGENCY OPERATING CONDITIONS	PROPELLER SPEED APPROX R.P.M.	GENERATOR														BLOWER MOTOR SWITCH	MOTOR			CONTROLLER					OVERLOAD RELAY	MS FORDONIAN DIESEL-ELECTRIC
		NO. 1.								NO. 2							No 1	No 2		POSITIONS			STARTING AND RESISTOR POINTS			
		SWITCHES			RES	INS'MT				SWITCHES			RES	INS'MT			SWITCHES		INS'T							
		CUT-OUT	SHUNT FIELD	SERIES FIELD	RHEOSTAT	VOLTMETER	AMMETER	EXCITATION SWITCH	AUXILIARY SWITCH	CUT-OUT	SHUNT FIELD	SERIES FIELD	RHEOSTAT	VOLTMETER	AMMETER		CUT-OUT	CUT-OUT	AMPERES FIELD	REVERSE	OFF	RESET	STARTING POINTS 1 to 5	RESISTOR POINTS 6 to 17		
GEN. No 1 & MOT No 1	60 to 80	2	8	4	10/17	10/17	16	6		3	7	5	9	9		11	12	13	15	14	1	15	16		15	SEQUENCE OF OPERATION
EXCITATION FROM GEN No 2		U	C	C	Aj	Aj for Load	Rd	U		D	C	O	Aj	250 volts		C	U	D	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
GEN No 1 & MOT No 2	60 to 80	2	8	4	10/17	10/17	16	6		3	7	5	9	9		11	12	13	15	14	1	15	16		15	SEQUENCE OF OPERATION
EXCITATION FROM GEN No 2		U	C	C	Aj	Aj for Load	Rd	U		D	C	O	Aj	250 volts		C	D	U	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
GEN No 2 & MOT. No 1	60 to 80	2	7	4	9	9		6		3	8	5	10/17	10/17	16	11	12	13	15	14	1	15	16		15	SEQUENCE OF OPERATION
EXCITATION FROM GEN No 1		D	C	O	Aj	250 volts		D		U	C	C	Aj	Aj for Load	Rd	C	U	D	Rd.	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
GEN No 2 & MOT No 2	60 to 80	2	7	4	9	9		6		3	8	5	10/17	10/17	16	11	12	13	15	14	1	15	16		15	SEQUENCE OF OPERATION
EXCITATION FROM GEN No 1		D	C	O	Aj	250 volts		D		U	C	C	Aj	Aj for Load	Rd.	C	D	U	Rd.	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
GEN No. 1 & MOT. No. 1	60 to 80	2	7	8	10/17	10/17	16	4	5	3	6	9		Read excitation volts on this INST		11	12	13	15	14	1	15	16		15	SEQUENCE OF OPERATION
EXCITATION FROM AUX EXCITER		U	C	C	Aj	Aj for Load	Rd	U	U	DS	O	O				C	U	D	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
GEN. No. 1 & MOT No 2	60 to 80	2	7	8	10/17	10/17	16	4	5	3	6	9		Read excitation volts on this INST		11	12	13	15	14	1	15	16		15	SEQUENCE OF OPERATION
EXCITATION FROM AUX EXCITER		U	C	C	Aj	Aj for Load	Rd	U	U	DS	O	O				C	D	U	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
GEN. No. 2 & MOT No 1	60 to 80	2	6	9		Read excitation volts on this INST		4	5	3	7	8	10/17	10/17	16	11	12	13	15	14	1	15	16		15	SEQUENCE OF OPERATION
EXCITATION FROM AUX EXCITER		DS	O	O				D	D	U	C	C	Aj	Aj for Load	Rd	C	U	D	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS
GEN. No. 2 & MOT. No. 2	60 to 80	2	6	9		Read excitation volts on this INST		4	5	3	7	8	10/17	10/17	16	11	12	13	15	14	1	15	16		15	SEQUENCE OF OPERATION
EXCITATION FROM AUX EXCITER		DS	O	O				D	D	U	C	C	Aj	Aj for Load	Rd	C	D	U	Rd	Ah'd or As'n					C	SWITCH POSITIONS INSTRUMENT READINGS

NOTE:– U = SWITCH CLOSED IN THE "UP" POSITION

D = SWITCH CLOSED IN THE "DOWN" POSITION

AH'D = AHEAD AS'N = ASTERN

Rd = READ INSTRUMENT WHEN MOVING CONTROLLER HANDLE

DS = DISCONNECT STRAP ON BACK OF PANEL AND THROW SWITCH "DOWN"

C = SWITCH CLOSED

O = SWITCH OPEN

Aj = ADJUST FOR VOLTAGE

To Operate With Both Generators at Speeds Between 90 and 120 R.P.M.: In order to obtain speeds between 90 and 120 R.P.M. the same procedure is followed as when operating with both generators and motors at a motor speed of 120 R.P.M., the only exception being that it is necessary to weaken the field of the generator not supplying current for excitation and auxiliary purposes by cutting-in the field rheostat for speeds from 120 R.P.M. down to 105 R.P.M. To obtain a speed range of 90 to 105 R.P.M. it is necessary to close the series field switch on this generator in addition to cutting in the field rheostat as mentioned above.

Emergency Operation

To Operate With Both Generators With One Motor Armature Cut Out: In case one of the motors has been disabled it can be cut out of the circuit by throwing its corresponding switch to the "down" position. In this case only one generator is used to supply line current to the motor armature while the other generator is used to supply excitation and control current to the motor and generator fields and the master controller and current to the auxiliary panel.

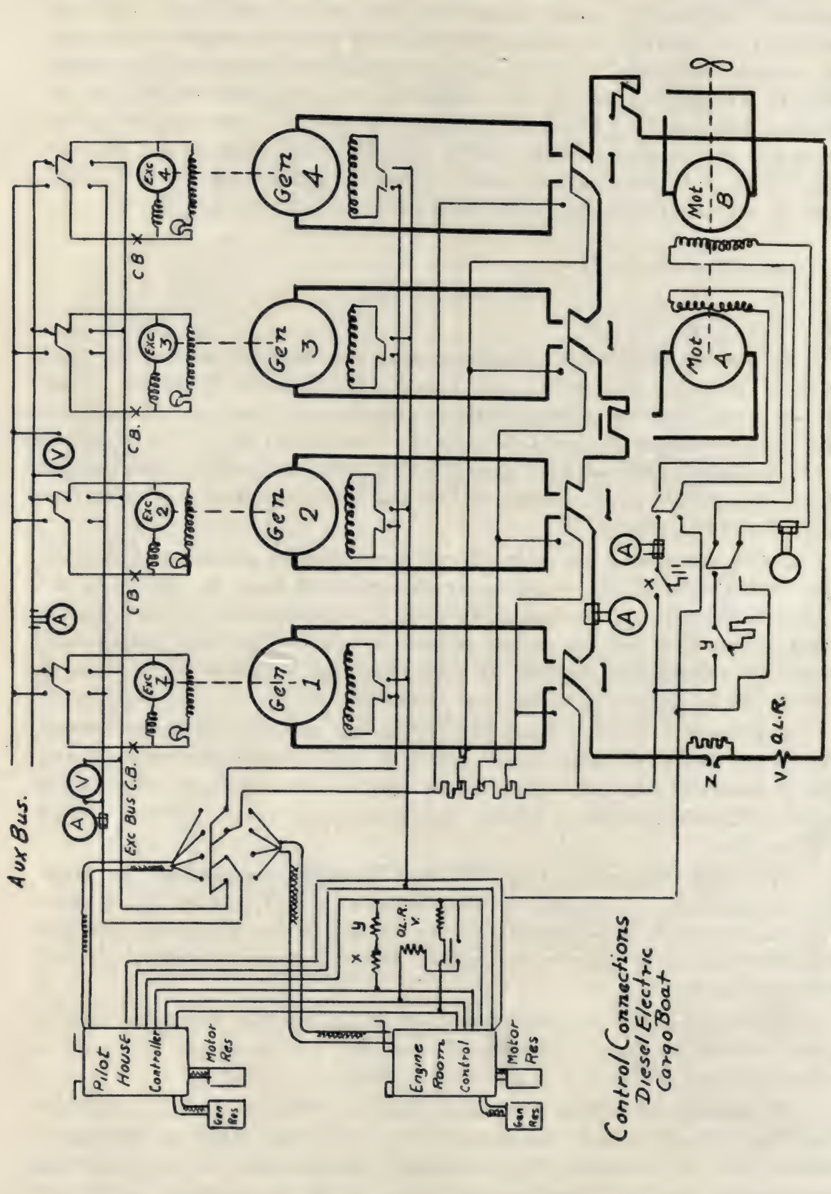
If generator No. 1 has been chosen for excitation purposes, the excitation switch and also No. 1 generator cutout switch must be thrown to the "down" position. At the same time No. 2 generator series field switch must be closed and its cutout switch thrown to the "up" position in order to supply line current to the motor still operative and whose switch should be closed in the "up" position.

If generator No. 2 is used for excitation purposes, its cutout switch should be thrown to the "down" position, and the excitation switch and No. 1 generator cut-out switch to the "up" position and the series field switch of generator No. 1 closed. Both generator field switches should be closed.

The field rheostat of the generator supplying current to the armature of the motor should be cut in to obtain about 175 volts before starting and then cut out gradually so as to obtain the maximum possible speed of the ship without exceeding the maximum allowable current through the motor.

250 volts should be maintained on the generator supplying excitation current, as this is the normal voltage for operating the control group and for giving full motor field current.

To Operate With One Generator Only and With One Motor Armature Cut Out: In this case, the procedure will be the same as that given above, with the exception that excitation current must be supplied from an external source, which in this installation is the auxiliary generator. The proper connections should be made on the auxiliary panel to connect with the line to the auxiliary switch on the control panel. **Disconnect the strap on the back of the panel and in the line of the generator not being used.** Both the auxiliary and excitation switches can then be



Control Connections, Diesel Electric Cargo Boat. – (General Electric System).

thrown either up or down, depending on which generator is to be used. If generator No. 1 is to be used the auxiliary and excitation switches should be thrown up or on the side marked generator No. 2. This connects the supply from the auxiliary panel to the excitation circuits and also places the voltmeter of generator No. 2 across this supply. The cut-out switch of generator No. 1 should be closed in the "up" position and that of generator No. 2 in the "down" position. Thus the voltage of the supply current to the motor armature will be registered on generator No. 1 voltmeter, while the voltage of the auxiliary generator for excitation purposes will be indicated on the voltmeter of generator No. 2. The field switch of generator No. 1 should be closed, while that of generator No. 2 should be left open.

If generator No. 2 be used in place of generator No. 1 the operation should be reversed, **making certain that the strap in the line of generator No. 1 has been opened and that of generator No. 2 closed.**

The auxiliary and excitation switches should both be thrown down or on the side marked generator No. 1. The cut-out switch of generator No. 2 should be closed in the "up" position and that of generator No. 1 in the "down" position. The field switch of generator No. 2 should be closed while that of No. 1 should be open. The voltage of the motor armature current will be registered on the voltmeter of generator No. 2, while the voltage of the auxiliary generator will be indicated on the voltmeter for generator No. 1.

Since the auxiliary generator is rated at 15 kw., and since the power required for excitation of one generator and one motor field is about 7.5 kw., and that required for the blower motor about 5.5 kw., there is only 2 kw. remaining for the auxiliary pumps and lights. It will, therefore, be necessary to devise some method whereby the necessary auxiliaries may be kept in operation. If it is decided to shut down the blower motor intermittently to relieve the auxiliary generator, the temperature of the motor should be carefully watched for overheating.

Operating Cautions

In the course of operation the following cautions should be observed:

1. Never open the generator or motor line switches until the controller handle has been placed in the "off" position and the main line and motor field contactors thus opened.

2. Before opening the generator shunt field switches, cut in the resistance of the corresponding field rheostat.

3. Before changing the position of the control and excitation switch always open the shunt field switches as explained in paragraph 2.

4. If one motor alone is used, only one generator should be used to supply line current to this motor.

5. Do not run on the first four points of the controller to obtain speed changes for any length of time greater than that specified under

the heading, "Starting Resistor," as this will overheat the grids with a tendency to burn them out.

6. When operating under normal conditions with both generators running in series with both motors, the oil engine driving the generator not supplying excitation current should not be shut down until the controller is moved to the "off" position, and its respective generator switch thrown down to the "cut-out" position, otherwise the unit shut down will tend to run as a motor in the opposite direction from current supplied by the other generator.

If the oil engine which drives the generator supplying excitation current is shut down, the line current will fall off due to lack of excitation. and the motors will stop.

It is therefore evident from the above conditions that the controller handle should always be moved to the "off" position before taking a generating unit off the line.

Maintenance of the Control Equipment

The frequency and thoroughness of inspection necessary to keep the control equipment in the best of operation depends largely on the operating conditions. While casual inspections necessarily follow from one's interest in the machinery it is very essential for consistent operation that a thorough inspection be made before each trip.

The inspection of the various parts can best be made by operating the machinery and then noting any faults.

Operating Test

At each inspection, the control group should be operated by means of the master controller. To do this the single pole double throw line switches of the generator should be open and the generator field switches closed and rheostats adjusted to give 250 volts on each generator so that the master controller may be operated first with one generator and then the other supplying the control current.

The controller handle should be turned from point to point and on the reset position it should be noticed that the overload relay resets and that on the next position or first point, that the ahead or astern contactors as selected, close properly. On points 2, 3, 4 and 5 the resistance contactors should close in the proper sequence and they should operate at about the same speed.

Inspection

At each inspection all parts of apparatus should be examined, cleaned, adjusted, or repaired if necessary, and the following points should be observed as follows:

Master Controller: (a) Inspect for weak fingers, imperfect contact, and loose connections.

(b) When dirty, clean the contacts and apply a small quantity of lubricating oil to the contacts with a piece of cheese cloth.

Engine Room Panel: (a) Inspect for loose connections.

(b) Examine the knife edges of the switches and see that good contact is maintained.

Control Group: The cable connections to the contactors and overload relay should be examined to insure that they are tight.

The contactors and relay of the control group should be inspected and properly maintained by observing the following:

(a) Examine the contact tips and tighten the screws holding them if loose.

(b) Renew contact tips when worn half way through.

(c) When renewing a contact tip, if the surface against which it rests has become rough or pitted, due to poor contact from a loose screw or similar cause, it should be made smooth.

(d) The contact tips close with a butting and rolling movement which tends to remove any roughness caused by arcing. If for any reason the tips become extremely rough, they should be filed smooth.

(e) The screws holding the contactors in place should be examined to see that they are tight.

(f) The pigtail shunts should be examined for wear and breakage.

(g) Examine the arc chute sides. When they are one-half burned through, they should be replaced by new ones.

(h) Inspect for loose or missing nuts and screws, broken or not split cotter pins, and broken wipe springs.

Overload Relay: (a) Keep the contact points clean.

(b) Trip the relay occasionally and see that the armature moves freely.

RECOMMENDED PROCEDURE FOR DRYING OUT AND TESTING THE INSULATION RESISTANCE OF THE GENERATOR AND MOTOR WINDINGS

Insulation Resistance

Although the generators and motors are provided with waterproof insulation, this insulation should be kept dry, and after standing for a long time, either during the period of installation or during interruption of service after installation, the insulation and also other parts are likely to become covered with moisture to a certain extent, and possibly the terminals or other exposed parts may become dirty, which will tend to reduce the insulation resistance and make it unsafe to operate at the

full potential. In order to determine the condition of the insulation, its resistance should be taken.

The insulation resistance can be readily measured by the use of a megger, which gives resistance in megohms. When this is not available the insulation resistance may be measured by means of a high resistance direct current voltmeter. The deflection of the meter is directly proportional to the current flowing through it and inversely proportional to the resistance of the circuit with constant potential across it.

In this installation the auxiliary generator can be used to supply a 250-volt circuit to measure the insulation resistance of the main generators and the motors.

To test for insulation resistance, attach the voltmeter directly across the terminals of the supply line and note the reading. Never use a voltmeter whose scale registers lower than the voltage supplied. The resistance to be measured is next connected in series with the voltmeter and a second reading taken and noted.

The resistance "X" is then given by the formula:

$$\frac{Rm}{Rm \div X} = \frac{Dm}{V} \text{ whence } Rm \div X = \frac{VRm}{Dm}$$

Where Rm = the Resistance of the voltmeter used

Dm = the deflection of the voltmeter with the resistance in series

V = the voltage supply when taking the reading Dm

X = the insulation resistance sought.

In making this test first determine whether either side of the circuit is grounded or has a low insulation resistance by connecting the voltmeter between each side of the supply circuit and ground. Next connect the line of lower insulation resistance to the frame of the motor or generator to be tested through a fuse of 5 or 10 amperes capacity or a resistor of from 100 to 500 ohms such as an incandescent lamp. Failure to observe this precaution may result in personal injury. One terminal of the voltmeter should now be connected to the other line, and the remaining terminal of the voltmeter connected first to the frame of the machine to be tested to determine the voltage of the line V and then transferred to the winding whose insulation is to be tested. The resistance of the insulation will then be in series with the resistance of the voltmeter and the reading obtained when so connected will be small when the resistance of the insulation is high.

As a conservative value the insulation resistance in megohms should not be less than ½ megohm.

To Dry Out the Windings

When the insulation resistance is found to be below the amount previously specified, the windings should be dried out after determining that all exposed terminals and connections are thoroughly cleaned, as leakage may be due to dirty connections as well as to moist insulation.

The following procedure is recommended to dry out the windings electrically:

1. Tie the boat securely to the dock.

2. Prepare to operate with one generator and both motors and with excitation supplied from the auxiliary exciter. The operations covering the above conditions are shown by the last four columns of operation chart No. 2 with the exception that both motor cut-out switches are closed in the up position.

3. Select the generator to be dried out first and turn its shunt field rheostat all the way in before closing the shunt field switch.

4. If generator No. 1 is to be used, refer to either the fifth or sixth columns of chart No. 2, making the proper motor switch connections as stated in paragraph (2). If generator No. 2 is to be used refer to either the seventh or eighth column and throw both motor switches "up" as stated above.

5. Note the generator voltage which should not exceed approximately 25 per cent normal. It may be necessary to reduce the excitation voltage of the auxiliary exciter sufficient to obtain this condition, but the voltage should not be dropped below a point sufficient to operate or keep the contactors closed.

6. Move the reverse handle of the controller ahead or astern, depending on the position of the boat in the dock.

7. Follow the sequence of operation as shown in chart No. 2 and note carefully the line and field amperes recorded on each of the five starting resistor points so as not to exceed full load current. Note also the propeller speed so as not to exceed a value which might break away the moorings.

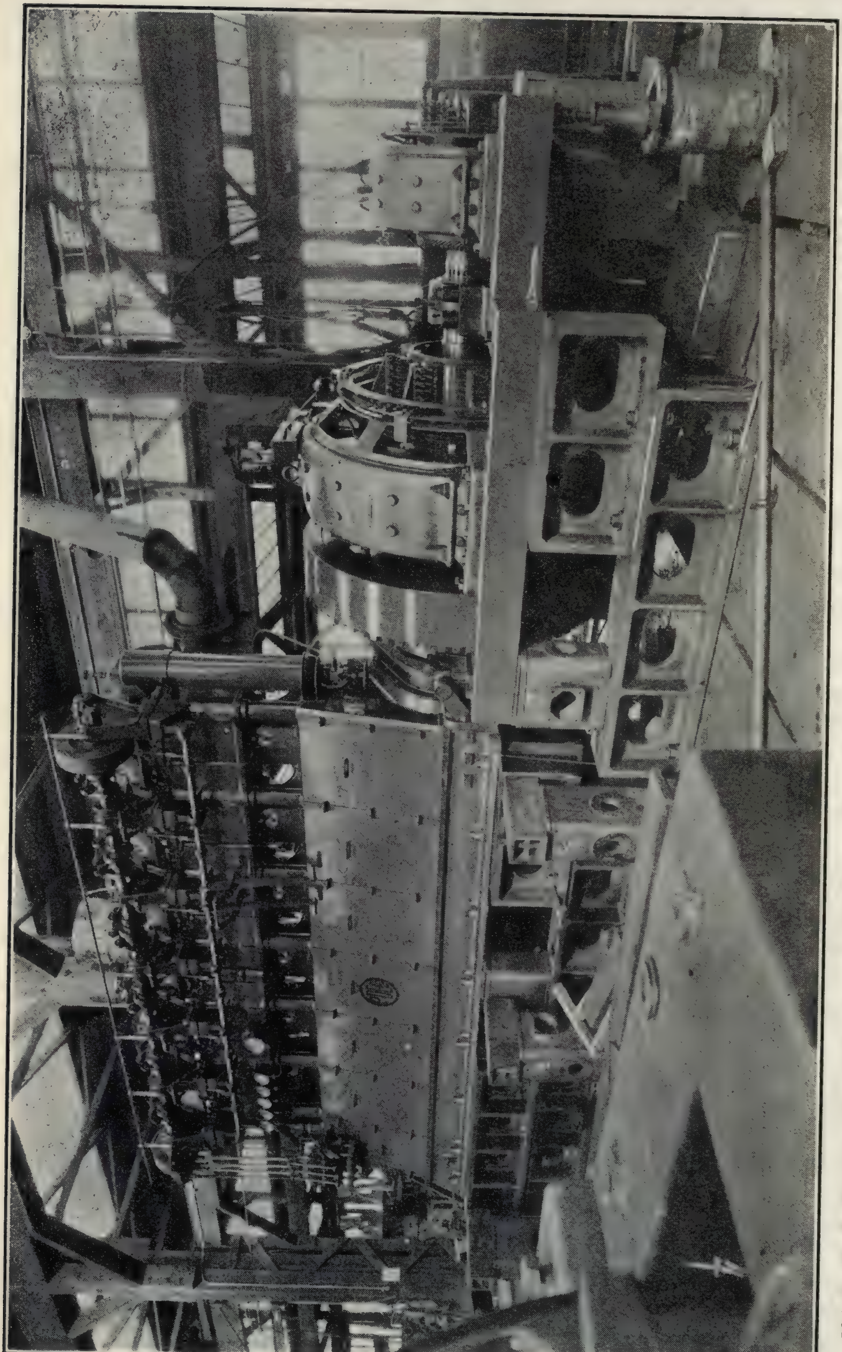
8. Adjust the generator field and exciter field if necessary, so as to obtain the maximum line current consistent with the above conditions.

9. The temperature of the windings should not attain 85 degrees C. (185 degrees F.) in less than two hours, and should never exceed this temperature during the drying out process.

10. Continue this drying out process until the insulation resistance is equal to or greater than the amount previously given.

11. After one generator is dried out, repeat the operation with the other generator, following the same method as given above.

12. While various methods of drying out might be possible with this apparatus, no method should be used in which the armature is not allowed to revolve, since local heating at the brushes might cause warping of the commutator.



A 525 B. H. P. Pacific Diesel-Werkspoor Engine on Test in Shop of the Pacific Diesel Engine Co., Oakland, California, Prior to Being Installed in a Golden Gate Ferry Co.'s Ferry at San Francisco Harbor. General Electric Motor Installation.

OUTPUT																					CONSUMPTION					
Main Engine																				Mechanical Eff in %	Fuel Oil		Thermal Eff. % 2545/HV·FC	Lubricating Oil		
Time	Rev Counter	R.P.M.	B.H.P.	Developed K.W.	Indicated Horse Power from Cards																Lbs per Hour	Lbs per B.H.P. Hour		Amount in Circ	Amount left in System	Make up during Test
					Cyl 1		Cyl 2		Cyl 3		Cyl 4		Cyl 5		Cyl. 6		Total	Average	Average							
					MEP	IHP	MEP	IHP	M.E.P.	IHP	M.E.P.	I.H.P.	M.E.P.	I.H.P.	M.E.P.	I.H.P.	I.H.P.	M.E.P.	I.H.P.							
2 33	15230		540	402 2																				Start 60.5		
3 33	28850	227	520	388.2	76	90.5	103.5	123	81	96.5	88.5	105 5	91	108	95	113	636.5	89.2	106	81.7						
4 33	42410	226	474	353 8	72	85.5	91.5	105	85	101	76	90	80	95	85	101	581 0	81.6	96.2	81.7	205	432	31.1			1- Gal Hand Lubrication
5.33	55 910	225	517	385.8	91	107	92	108 5	86.5	102	90	106	93.5	110	97	114.5	648.0	91.6	108	798	Filling Tank					
6.33	69470	226	508	379 6	79	93 5	98	116	75	89	83.5	99	93.5	110.5	102.5	121.5	629.5	88.6	104.9	80.8	229	.450	29.9			
7 33	82910	224	508	379.6	78	91.5	99.5	117	81	95.5	90.5	106	86	101	100	117.5	628.5	89.2	104.7	81.0	223	.438	30.7			
8.33	96470	226	557	415 5	85	101	100	118.5	85	101	106	125.5	89	105.5	101	119.5	671.0	94.3	112.0	83.0	221	.409	32.8			
9.33	110120	227 5	558	416 0	108	128 5	102	121 5	73 5	87.5	98	116.5	79.5	95	101	120	669.0	93.7	110.7	83.5	228	.392	34.2			
10.33	123650	225.5	560	417 5	95	112	102	120 5	95	112	91.5	108	96	101.5	99	117	671.0	94.7	112.0	83.5	219	.392	34.2			
11.33	137210	226	536	400 0	90.5	107	96	109	76.5	91	91	108	93	110	97	115	640.0	90.7	107.0	83.7	Filling Tank					
12 33	150770	226	540	403 0	94.5	112	94.5	112	85	1.01	79.5	94.5	103	122	89.5	106	647.5	91.0	108.0	83.5	212	.393	34 1			
1 33	164330	226	540	403 0	92 5	110	94.5	112	84	99.5	87.5	103.5	95	112.5	93	110.5	648.0	91.0	108.0	83.5	212	.392	34.2			
2.33	177890	226	556	415 0																	217	.391	34.3		Fin. 49.5	
																								12-Gals. Total		
Average		226	532	396.9	87 4	104	97 6	115	82 5	97.8	89.3	105 7	90	106.5	96.4	114	643	90.5	107.0	82.3	218	.410	32 8	1-Gal. per Hour		

MAIN ENGINES

Rated B H P	525	Stroke	23 5/8
Rated R P M	225	Air Comp Stages	3
Rated M E P		Dia L.P.= I.P.= H.P.=	
Bore	15"	Stroke	

Test Log No. 1 of Official Shop Test of Engine for Golden Gate Ferry Co., by Pacific Diesel Engine Co., Oakland, California

PRESSURE

Main Engine Cyl 1 Comp Press	Cyl 1 Exh Press	Cyl 2 Comp Press	Cyl 2 Exh. Press	Cyl 3 Comp. Press	Cyl 3 Exh	Cyl. 4 Comp	Cyl. 4 Exh	Cyl 5 Comp. Press	Cyl 5 Exh. Press	Cyl 6 Comp Press	Cyl 6 Exh Press	Compressor H.P.	Compressor I P	Compressor L P.	Lubricating Oil
												1050	295	42	4.5
500	51	490	50	510	51	520	52	500	51	490	50	1070	300	42	4.5
500	51	490	50	510	51	520	62	500	51	490	50	1040	270	37	4.5
500	51	490	50	510	51	520	52	500	51	490	50	1050	270	37	4.5
500	51	490	50	510	51	520	52	500	51	490	50	1100	275	37	4.5
500	51	490	50	510	51	520	52	500	51	490	50	1150	280	37	4.5
500	51	490	50	510	51	520	52	500	51	490	50	1120	265	33	4.5
500	51	490	50	510	51	520	52	500	51	490	50	1120	260	33	4.0
500	51	490	50	510	51	520	52	500	51	490	50	1120	260	33	4.0
500	51	490	50	510	51	520	52	500	51	490	50	1110	260	33	4.0
500	51	490	50		51	520	52	500	51	490	50	1120	265	33	4.0
500	51	490	50		51	52	52	500	51	490	50	1120	265	33	4.0
												1110	265	33	4.0
500	51	490	50	510	51	520	52	500	51	490	50	1116	272	35.6	4.3

TEMPERATURE / **MISCELLANEOUS**

Main Engine Exhaust Temperature Cyl 1	Cyl 2	Cyl 3	Cyl 4	Cyl 5	Cyl 6	Circulating Water Inlet	Circulating Water Outlet	Circulating Water Gals per Hour	Gals per B.H.P. Hour	Gen. Volts	Gen. Amps	Exct Volts	Exct Amps
620	790	730	720	710	690	53	138	2290	4.15	245	1500	90	40
610	820	750	720	710	680	53	133	2230	4.30	245	1450	90	40
530	750	706	650	650	630	53	131	2220	4.68	240	1350	85	38
520	760	760	610	700	520	53	136	2230	4.32	240	1470	85	38
580	630	740	590	740	630	53	136	2210	4.35	240	1458	80	38
520	680	730	610	670	650	53	136	2230	4.38	240	1450	80	38
580	750	770	740	740	710	53	138	2245	4.04	250	1520	95	40
860	840	790	780	770	610	53	138	2225	4.00	250	1520	100	42
780	780	740	730	740	720	53	136	2230	4.00	250	1525	100	42
790	790	760	760	750	740	53	134	2230	4.15	250	1460	100	42
770	790	750	740	740	750	53	134	2230	4.13	250	1470	100	42
790	760	790	730	760	730	53	134	2230	4.13	250	1470	100	42
780	750	790	740	790	720	53	136	2230	4.01	250	1515	95	42
672	761	754	719	728	684	53	135.5	2231	4.20	248	1486	101	403

FUEL OIL

SP GRAVITY	95
GRAVITY BAUME	18°
APPROXIMATE B.T.U	18970

Test Log No. 2 of Official Shop Test of Engine for Golden Gate Ferry Co., by Pacific Diesel Engine Co., Oakland, California

CAUSES AND REMEDY OF FAULTS IN DYNAMOS

Failure to Built Up: A number of causes may prevent the field magnets from attaining their full strength, which is commonly the reason of failure to built up. A primary cause is absence of residual magnetism which is shown by a piece of iron not being attracted when brought near the pole-pieces when the dynamo is not running. This may be remedied by exciting the field coils from an outside source or, if the demagnetization is due to the influence of an adjacent machine one of the machines should be turned half way around or the magnetism of the poles reversed.

If the field coils are connected so as to oppose each other, there will be no resultant magnetism and, therefore, the dynamo cannot generate its full E.M.F. Nearly all machines have successive poles of opposite polarity, hence a fault of this nature can be detected by testing the polarity by means of a compass needle. If the same end of the needle is attracted by successive poles it shows that they oppose each other. In such a case the connections of the field coils requiring a reversal of polarity should be changed.

An open circuit in the machine will, of course, prevent the flow of current and, therefore, the field magnetism will remain weak. This fault may be located by a careful inspection of the machine or by testing the parts separately by the fall of the potential method. A short circuit in the machine will also prevent it from attaining its full E.M.F. Such a fault may be located in the same way as an open circuit.

If the connections are such that the current generated by the armature and flowing through the field coils opposes the residual magnetism, the latter will be destroyed and no E.M.F. generated. A reversal of residual magnetism will not remedy the fault, as the direction of the current generated by the armature will also be changed and destroy the magnetism as before. This trouble may be overcome by changing the direction of the armature, or by reversing **either** the armature or field connections. An incorrect position of the brushes may prevent a dynamo from attaining its full E.M.F. In the case of a shunt dynamo, building up may be prevented by a short circuit in the external circuit or by having too great a load in starting. In the case of a series dynamo, building up may be prevented by the resistance of the external circuit being too great.

Sparking: This is a common trouble and, owing to the large number of causes to which it may be due, it is one of the most difficult to locate.

An incorrect position of the brushes will cause sparking, but this is readily overcome by shifting them to their proper position. It may also be due to the brushes not being an equal distance apart along the circumference of the commutator segments between the forward edges of successive sets of the brushes, which number should be the same in every case.

A cause of sparking which is evident upon inspection is a rough or uneven commutator, or poor brush contact. A small amount of roughness in the commutator may be eliminated by use of a fine file or sandpaper. Emery cloth should never be used for this purpose, as the emery particles remain in the metal and cut both commutator and brushes. If the commutator is very rough, uneven, or eccentric, it should be carefully turned down in a lathe and smoothed off. Small particles of copper which may have become lodged in the mica insulation during the process should be picked out. Poor brush contact may also be due to an uneven brush or the presence of dirt. A carbon brush can be beveled to fit the commutator by placing it in its position and inserting a strip of sand paper between it and the commutator and drawing it back and forth, face against the brush. The brush then receives the same curvature as the commutator.

Sparking will result from an excessive flow of current, whether this excess is either continuous or intermittent. Overloading of a machine or a line short-circuit may cause continuous sparking. This cause will be evident by the excessive heating of the whole armature. A short circuit, or a grounded circuit in the armature will cause an excessive current and, consequently, sparking. These faults may be located by the fall of potential method. In order to have a complete short-circuit within the machine, two grounds are necessary.

If the field magnetism is weak the armature will cause more than its normal amount of distortion of field, and sparking will result. A condition of this nature may be brought about by a short-circuit, open circuit, or grounds in the field coils, as well as by an incorrect connecting up of the coils.

ADVANTAGE AND DISADVANTAGES OF SHUNT MOTORS

The great advantage of shunt motors is that their speed is practically constant.

The disadvantages are as follows:

1. The torque increases only in proportion to the armature current since the field strength is constant.

2. There is a high potential between the terminals of the field winding.

3. Opening the field circuit suddenly causes a large spark and a high potential due to self induction.

4. The many turns of fine wire of the field coils involve extra expense since fine wire costs more per pound than coarse, the labor of winding is considerable, and a large amount of insulation is necessary.

5. Where shunt motors are run intermittently it is customary to keep the field coils constantly charged, the motor being started and stopped by closing and opening the armature circuit through the resistance. Keeping the fields charged when the motor is not in use is, of course, a loss of energy.

Uses of Shunt Motors: A constant speed is of the greatest importance in some classes of work, and this advantage makes the use of shunt motors very extended.

Shunt motors are used in machine shops, factories, and to run printing presses, pumps, elevators, etc., or in general where good speed regulation is required.

CAPACITY OF MOTOR FOR PULLING

As a guide for determining the maximum depth of well at which a motor of a given rating can safely be installed for pulling work, the following formula is of much service. It is based on the maximum torque of the motor, but has been found sufficiently conservative so that the motor heating will normally not be excessive under the usual operating conditions.

$$\text{Maximum depth of well} = \frac{R \times E \times L \times K}{w \times d}$$

in which

R = radio of motor speed to corresponding bull-wheel speed

E = mechanical efficiency of rig (usually varying from 0.5 to 0.7)

L = number of load lines used in the tackle for pulling the tubes.

w = weight of tubing in lb. per foot.

d = diameter of bull-wheel shaft in inches.

K = a constant, depending upon the motor used.

The constant K is determined as follows:

$$K = \frac{1260 \times \text{H.P.} \times T}{\text{R. P. M.}}$$

in which,

H.P. = horsepower rating of motor on high speed.

T. = max. torque of motor in per cent of full load torque.

R.P.M. = full load high speed of motor.

The extreme condition which may be encountered is pulling rods and tubing together with the tubing full of oil. This may be taken into account by determining the total weight per foot of this load and using this figure for "w" in the formula.

ELECTRICAL DATA:

Volt: The practical unit of electrical pressure analogous to head or pressure in hydraulics.

Ampere: The practical unit of electrical strength or rate of flow of current. Analogous to rate of flow of water through a pipe in gallons per second.

Ohm: The unit of resistance. Analogous to the loss of head due to the flow of water in a pipe.

Coulomb: Unit of quantity = one ampere per second.

Volt = Ampere times Ohms.

Ampere = volts divided by Ohms.

Ohm = Volts divided by Amperes.

WORK AND POWER:

Work, or energy, is expended in a circuit or conductor when a current of electricity flows through it. The unit of electrical work or energy is called the Joule, after an eminent English scientist. If E is the electromotive force, or difference of potential, in volts that causes Q coulombs of electricity to flow through a circuit, the work expended in joules is $J = E \times Q$.

If an electromotive force, or difference of potential, of E volts causes a current of I amperes to flow for t seconds through a resistance of R ohms, then

$$J = E^1 t$$

$$J = \frac{E^2 t}{R}$$

$$J = I^2 R t$$

The joule may be defined as the work done when I ampere flows for I second through a resistance of I ohm.

The watt-hour is an extensively used unit of work. Watt-hours equal the product of the average number of watts and the number of hours during which they are expended. One kilowatt-hour = 1,000 watt-hours, or the product of the average number of kilowatts and the number of hours.

Power (P) which is the rate at which work is done, is equal to the work divided by the time, and may be calculated by one of the following formulas:

$$P = 1E = 1^2R = \frac{E^2}{R} = \frac{J}{t}$$

If I is in amperes, R in ohms, E in volts, J in joules, and t in seconds, P is in watts.

The watt, or unit of electric power, is equal to 1 joule per second. It is the rate at which work is expended when 1 ampere flows through a resistance of 1 ohm. The watt is too small a unit for convenient use in many cases, so that the kilowatt (KW.) or 1,000 watts is frequently used. 1 H.P. equals 746 watts; therefore

$$H.P. = \frac{P \text{ (in watts)}}{746} \text{ or, } H.P. = \frac{P \text{ (in kilowatts)}}{.746}$$

COEFFICIENTS OF LINEAR EXPANSION.

Material	For 1° Fahr.	For 1° Cent.
Aluminum	.0000128	.0000230
Brass	.0000055	.0000099
Brick, Fire	.0000049	.0000088
Bronze	.0000100	.0000180
Copper	.0000093	.0000167
Glass	.0000049	.0000088
Gold	.0000080	.0000144
Iron, Cast, Gray	.0000059	.0000106
Iron, Wrought	.0000063	.0000113
Lead	.0000162	.0000292
Mercury	.0000333	.0000600
Monel	.0000076	.0000137
Nickel	.0000071	.0000127
Platinum	.0000049	.0000088
Porcelain	.0000020	.0000036
Silver	.0000107	.0000193
Slate	.0000058	.0000104
Steel, Cast	.0000064	.0000115
Steel, Rolled	.0000056	.0000101
Tin	.0000124	.0000223
Zinc	.0000162	.0000292

MELTING POINTS.

Material	Fahr. Degrees	Cent. Degrees
Aluminum	1217	658
Brass	1643	895
Bronze	1823	995
Copper	1981	1083
Gold	1945	1063
Iron, Cast, Gray	2200-2300	1204-1260
Iron, Cast, White	2000-2100	1093-1149
Iron, Wrought	2732	1500
Lead	621	327
Mercury	—37.9	—38.8
Monel	2408	1360
Nickel	2646	1452
Platinum	3191	1755
Silver	1762	961
Steel, Mild	2687	1475
Steel, Hard	2588	1420
Tin	450	232
Zinc	786	419

DIMENSIONS, RESISTANCES AND SAFE CARRYING CAPACITY OF COPPER WIRES:

B. & S. Gauge No.	Diameter in Mils., or Thousandths of an inch	Area in Circular Mils.	Ohms per 1,000 ft.	—Safe Amperes— Rubber-covered	Weather-proof
----	1000	1,000,000	.01038	650	1000
----	894	800,000	.01297	550	840
----	775	600,000	.0173	450	680
----	707	500,000	.02076	400	600
----	632	400,000	.02596	325	500
----	548	300,000	.0346	275	400
0000	460	211,600	.04906	225	325
000	410	167,805	.06186	175	275
00	365	133,079	.07801	150	225
0	325	105,592	.0983	125	200
1	289	83,694	.1240	100	150
2	258	66,373	.1564	90	125
3	229	52,633	.1972	80	100
4	204	41,742	.2487	70	90
5	182	33,102	.3136	55	80
6	162	26,250	.3955	50	70
8	128	16,509	.6288	35	50
10	102	10,381	1	25	30
12	81	6,530	1.590	20	25
14	--	4,107	2.591	15	20
16	51	2,583	4.019	6	10
18	40	1,624	6.391	3	5

H.P., K.W., AND K.V.A.

The output or work done by an engine is mechanical power and is measured in horsepower (H.P.).

The output of an alternating current generator is electric current and it is measured in kilovolt amperes (K.V.A.).

The useful output of a power plant is electric power and it is measured in kilowatts (K.W.).

The K.W. output of standard plants is reduced somewhat at extremely high altitudes because the capacity of the engine is reduced on account of the rarefied air.

The K.V.A. output of an alternator or power plant can be figured as follows:

Single Phase:

$$\frac{\text{Volts} \times \text{Amperes}}{1000} = \text{K.V.A. (Kilovolt-amperes)}$$

Two Phase:

$$\frac{\underset{\text{Phase 1}}{(\text{Volts} \times \text{Amperes})} + \underset{\text{Phase 2}}{(\text{Volts} \times \text{Amperes})}}{1000} = \text{K.V.A.}$$

Three Phase:

$$\frac{\underset{\text{Phase 1}}{(\text{Volts} \times \text{Amperes})} + \underset{\text{Phase 2}}{(\text{Volts} \times \text{Amperes})} + \underset{\text{Phase 3}}{(\text{Volts} \times \text{Amperes})}}{1732} = \text{K.V.A.}$$

In an alternating current circuit, watts or kilowatts (K.W.) can be measured only by a wattmeter. They cannot be found by multiplying volts by amperes as in a direct current circuit.

Under some conditions, K. V. A. as found from volts and amperes by the above rules, and kilowatts as measured by a wattmeter on the same circuit, may be the same. Usually, however, the watts will be less than the K.V.A. If we find the watts are 80 per cent of the K.V.A. we say the "Power Factor" is 80 per cent, because only 80 per cent of the current indicated by the ampere meters is transmitting power. The part of the current that does not transmit power is called "Wattless Current," and but little power is consumed in producing it, so when the power factor is low there may be a large output in volts and amperes indicated by the switchboard instruments with a comparatively small horsepower load on the engine.

The reason for this is that the voltage of an alternating current is continually changing. It runs up to a high value and then down to zero and up to a high value again in the opposite direction. This happens 60 times a second if it is a sixty-cycle current. On account of a magnetic action called inductance, there is usually some current flowing in the circuit at the instant when the voltage is zero, and that part of the current does not transmit any power because for that moment volts times amperes are zero and there are no watts.

The power factor may be high at one time and low at another. It depends on the load and the amount consumed.

$$\text{Amperes} = 1 \quad \text{Ohms} = R \quad 1 = \frac{E}{R} = \frac{W}{E} \quad R = \frac{E}{1} = \frac{E^2}{W} = \frac{W}{1_2}$$

$$\text{Volts} = E \quad \text{Watts} = W \quad E = 1R = \frac{W}{I} \quad W = E1 = \frac{E^2}{R} = R1^2$$

To Determine the Size of Copper Wire for Any Given Service:

Let C. M. = Cir. Mils.

Let D. = Distance.

Let C. = Current.

Let L. = Loss in Volts.

21.5 is a "Constant" or figure always used.

$$\text{Then } \frac{\text{C.} \times \text{D.} \times 21.5}{\text{L.}} = \text{Cir. Mils.}$$

Example: It is required that 100 amperes be carried 350 feet on a 110-volt circuit, with a loss of 2 per cent in voltage. What is the cir. mils. required?

First, ascertain the loss in volts, or 2 per cent of 110 = 2.2 volts.

$$\frac{100 \times 350 \times 21.5}{2.2} = 337{,}500 \text{ cir. mils. or two No. 000 wires.}$$

Where a wiring table is not at hand and it is desired to ascertain the weight of any bare copper conductor, it can be roughly determined in accordance with the following:

One thousand feet of wire, having an area of 1000 circular mils, weighs approximately 3 pounds, and the weight of any bare conductor can, therefore, be determined by multiplying its area in circular mils by .003.

QUESTIONS A DIESEL ENGINE OPERATOR SHOULD BE ABLE TO ANSWER.

GENERAL SUBJECTS.

(1) Define the principle of operation of a Diesel engine.

(2) Define the Two-Stroke Cycle operation.

(3) Define the Four-Stroke Cycle operation.

(4) Explain the Four Events in a cycle.

(5) Why is a Diesel engine classified as a "constant pressure" engine?

(6) Explain the meaning of "adiabatic expansion."

(7) Explain the meaning of "isothermal expansion."

(8) What is meant by "thermal efficiency"?

(9) What is meant by "volumetric efficiency"?

(10) What is meant by "mean effective" of the engine?

(11) What is meant by "mean-indicated" pressure?

(12) What is meant by "mechanical efficiency" of the engine?

(13) What is meant by "thermo-dynamic law"?

(14) What is cavitation and how is it caused?

(15) What is a hydrokineter and for what purpose is it used?

(16) What is a dynamometer and for what purpose is it used?

(17) How is the horsepower of a Diesel engine ascertained?

(18) What is Brake Horsepower?

(19) What is Indicated Horsepower?

(20) What superiority has a Diesel engine over a steam engine for marine propulsion?

(21) For what is a clinometer used on shipboard?

(22) What is a planimeter?

(23) What is a fair fuel consumption per horsepower of a Diesel engine of 600 H.P.?

(24) How much heat temperature F. does 500 lbs. per square inch create?

(25) What is the principle on which an ordinary pyrometer works?

(26) What should a perfect vacuum be?

(27) Define atmospheric pressure and how much is it calculated per cubic foot?

(28) What is a pneumercator, and what is its purpose?

(29) What is the mechanical equivalent of a heat unit?

Note: The answers to following questions may be obtained by studying the subject matter in the different chapters of this book.

(30) What is meant by British Thermal Unit and how do you determine this measurement?

(31) What is meant by calorific value of fuel?

(32) Define the meaning of specific gravity.

(33) Define the meaning of viscosity.

(34) What is meant by coefficient?

(35) What is CO_2?

(36) What is the ideal percentage of CO_2 for efficient combustion?

(37) What are the principle constituents of fuel oil?

(38) What should be the ideal flashpoint for ordinary fuel oil?

(39) What is the meaning of Beaume or Twaddle degree measurement?

(40) What is the usual cause of spontaneous combustion in bunkers?

(41) What elements should be considered in certifying to the amount of fuel oil received, if the contract is by barrel, and payments are to be made by the ton?

(42) How many gallons are there to a barrel and how many barrels to a ton?

(43) What precautions should be taken before sending a man into a tank which has contained fuel oil?

(44) How are pressure gauges tested for accuracy?

(45) What is meant by scavenging efficiency in a cycle?

(46) What are the four maintenance principles upon which a Diesel engine operates?

(47) How is the fuel injected into the cylinder?

(48) Why is water-cooling necessary?

(49) How much pressure is necessary to supply the cylinders with fuel oil?

(50) What are the functions of injection devices?

(51) Name the necessary valves on Diesel engines.

(52) What are the functions the compressor performs?

(53) What are reservoirs of cylindrical forms called and what are they intended for?

(54) How is a compressor constructed?

(55) What is meant by stages on compressors, what advantages are secured?

(56) What is the object of a scavenging pump?

(57) What advantages, if any, are claimed for engines operating by the opposed piston principle?

(58) What necessitates high pressure in Diesel cylinders?

(59) Can scavenging be effected without valves?

(60) Explain the process of combustion in "heavy oil" engines.

(61) Explain the difference between the "trunk" type and cross-head piston.

(62) What advantages are claimed for "step" piston?

(63) Explain the working of "air-operated" piston valves.

(64) What is the method of actuating the valves?

(65) Explain how engines are timed.

(67) What are the usual methods of lubrication on Diesel engines?

(68) How are pistons on Diesel engines water cooled?

(69) How are injection air and fuel retarded?

(70) What is the usual method of removing the needle valve on fuel valve?

(71) Explain the construction of the usual types of valve attachments.

(72) What materials should be used for suction pipes in bilges?

(73) How many cubic inches are there in a gallon of oil?

(74) What is the average percentage of losses on a Diesel direct propelled ship, single, and how much on a twin propelled ship?

(75) What is the average percentage of electrical loss between generator and motor in an electrically driven Diesel ship?

(76) Why should not alternating current be used for the propulsion motors in an electrically driven Diesel ship?

(77) Explain the system of electrical propulsion on Diesel powered ships.

(78) Define the usual features to be found on fuel oil pumps.

(79) What is the theoretical lift of a pump?

(80) What is a Spray Preheater, for what purpose are they installed?

(81) Define a Spray Air Cooler.

(82) Define the different methods of oil filtering systems.

(83) Define an apparatus for re-cooling lubricating oil on Diesel plants.

(84) What material is mostly used to line stern bearings?

(85) What are the four qualifications a good lubricant should possess?

(86) What are the necessary constituents tar-oil should possess when used for fuel purpose on Diesel engines?

(87) What are oils classified as hydro-carbons?

(88) What effect will asphalt percentage to a large extent have on engine?

(89) What effect will sulphur percentage in fuel have on engine?

(90) What effect will an excessive amount of water in fuel have on engine?

(91) What importance has "paraffine content" in fuel?

(92) What effect will an undue amount of "ash" have in fuel?

(93) What is the "critical" point of an oil?

(94) What is the "deadweight" tonnage of a ship?

(95) What is the slip of a propeller?

(96) What is the pitch of a propeller?

(97) Explain the object of the thrust bearing.

(98) Explain the principle of "semi-Diesel" engines.

(99) What are closed and what are open nozzle fuel injection devices?

(100) Explain the principle of the "Sperry Compound Engine".

CHAPTER XII.

LOW COMPRESSION ENGINES.

HEAVY DUTY OIL ENGINES, MARINE AND STATIONARY LOW COMPRESSION

The low compression engine, generally termed the semi-Diesel engine, has the distinction that it operates under pressures up to about 250 lbs. per square inch. In construction it is far simpler and requires less knowledge than the high compression or usually known as the "full Diesel" engine. It follows in principle of construction the two-stroke cycle system. While there are manufacturers who are adhering to the four-stroke cycle, low compression engine, it must be agreed that the two-cycle in this respect is universally considered the ideal construction.

Low Compression Engine Pioneer of All Internal Combustion Engines: The statement made that the low compression engines are all modifications of the Hornsby-Akroyd engine must be disputed. The modern gas engine as well as the Diesel types are in reality an outcome of experiments made in the early sixties with the surface ignition system. Patents have been granted in the United States as well as foreign countries to inventors creating devices for power production through the methods of surface ignition. It was only after the electrical age began to be felt, that the modern gasoline driven engine was brought to the front. Thanks to the great German inventor, Dr. Rudolph Diesel, the Diesel received its marked attention.

Theory of Combustion: Oils of heavy viscosity will not ignite in the presence of air not sufficiently high in temperature. On the other hand, if the oil strikes a hot surface it will break up into hydro-carbons of minute particles, and, when assisted by an existing high temperature in the cylinder, it will materially be brought into useful form. The initial step in starting the engine is in modern types performed with the assistance of electrical starters receiving current of electricity from a battery, or, in some cases the hot bulb, hot pin, etc., is used. While the fuel, coming in contact on its entering the combustion chamber, with the hot tube or hot plate, or as previously stated, electrical device, the "kracking" of the fuel is accomplished before the piston reaches its dead center.

It was customary in years gone by to use a lighter grade of oils for the use of semi-Diesel engines, but of late the engines having been brought to a higher stage of perfection, and, as a matter of fact, any kind of fuel oil may be used in most standard types. Some manufactur-

ers of semi-Diesel engines are entirely ignoring the use of higher gravities of oils and are recommending the lower grades.

Semi-Diesel Engine A Factor of Importance on Land As Well As Marine: When considering the fact that the semi-Diesel engine has been recognized by the agricultural population, industrial and marine service as a factor of necessity in the welfare of the nation, the enormity of numbers in use will substantiate this statement. Not alone from the standpoint of economy, but also from the indisputable fact that the engine is the simplest mechanism among power generators, the adoption of this type has been exceedingly rapid. In earlier years much trouble was experienced with semi-Diesel engines, such as cracking of cylinders, pre-ignition, etc., which has been universally solved by the experiments made to create the highest type possible. In particular, pre-ignition, which, in some instances, was overcome by the use of water injection. With modern designs all serious troubles are entirely eliminated and the vertical as well as the horizontal semi-Diesel engine are highly satisfactory.

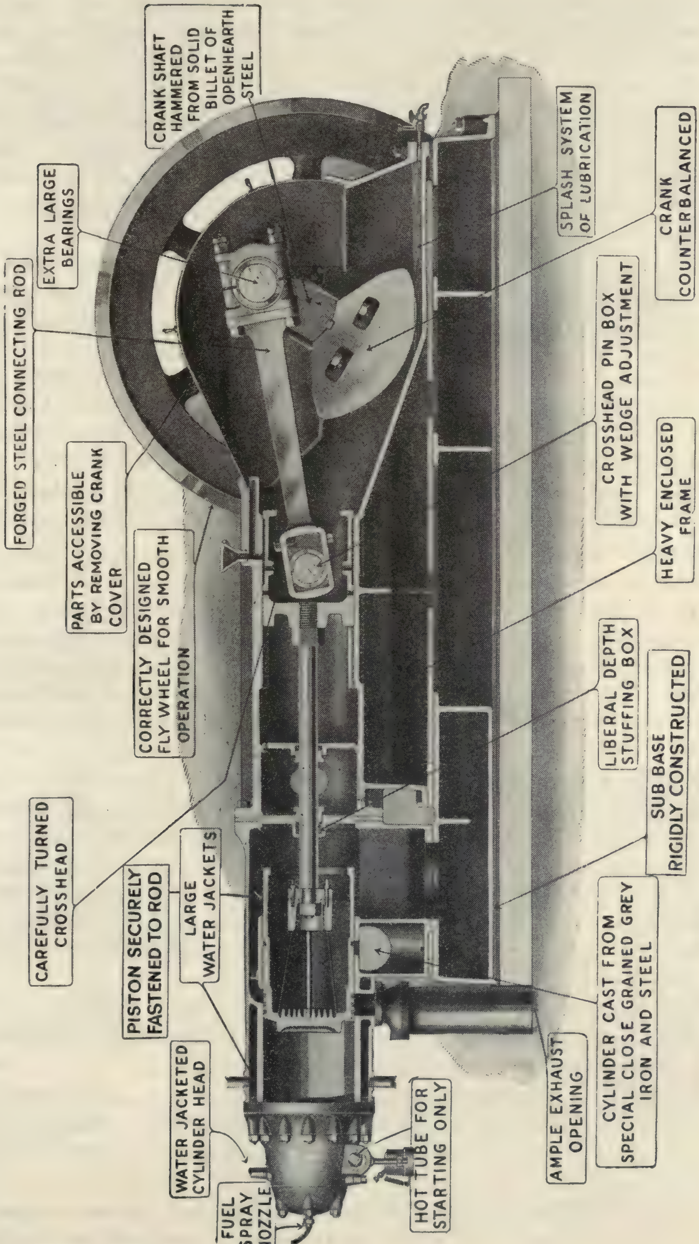
VITAL POINTS IN OIL ENGINE DESIGNS

In the designing of oil engines, of either the vertical or the horizontal types, many factors have to be considered. Where the engine follows the principle of high compression, a provisioncy must be made assuring the compact design of the pump delivering the oil against existing pressure in the cylinder, a moderate system of accomplishing the injection in the combustion chamber, ample water-cooling, and such means of lubrication as tend to minimize high temperatures, thereby avoiding carbonization of the lubricating oil and escaping the attendant complications.

As will be seen, when studying the different types of engines illustrated in this book, different methods are employed in most every respective make. To exemplify this, we will give a procedure to be found on the Chicago Pneumatic Tool Company's Giant Oil Engine.

This engine differs from all others in one or more of three broad features of design: The horizontal type of the engines, the use of a crosshead, and the use of a hot liner in the combustion chamber as a means of igniting the fuel, instead of a hot ball, hot bulb or electric ignition.

Horizontal vs. Vertical Construction: As previously explained, the mechanism of a vertical engine is rather a disadvantage in so far as accessibility is concerned; for all parts in the crank case must be reached through small openings. We do not mean to imply that there are not features in the vertical engine of very desirable nature, but rather to draw the attention of certain matters dealing with the practical features in general use. If it becomes necessary to remove the piston, the connecting rod must be diconnected from the crank pin, the cylinder head removed, and the piston drawn out of the cylinder by means of a chain



Sectional View Showing Parts of Giant Semi-Diesel Oil Engine—Class A-02.

block or some form of hoist. To remove the crankshaft, one or both flywheels must be taken off, the flanges which support the main bearings removed, and the shaft taken out of the frame endwise; all of which requires considerable extra floor space, especially in the case of engines direct connected to electric generators.

In horizontal engines matters already explained as causing extra time are minimized and other, rather undesirable features for stationary purposes overcome. The horizontal engine is more desirable for stationary power production, while the vertical engine is far preferable for marine work.

Crosshead vs. No Crosshead: In any two-stroke cycle engine not fitted with a crosshead, the crankcase must be as nearly air-tight as possible. The air for scavenging the cylinder must be compressed in the crankcase, and if it is not tight, will leak out and impair the scavenging, preventing efficient operation of the engine. This is so important that some builders put stuffing boxes on the outer ends of the main bearings. All the crank case covers are necessarily small and are bolted down on gaskets. This makes the parts within the case very inaccessible.

The design of Giant Engines enables the air for scavenging to be compressed in the crank end of the cylinder, and an air-tight crankcase is therefore unnecessary.

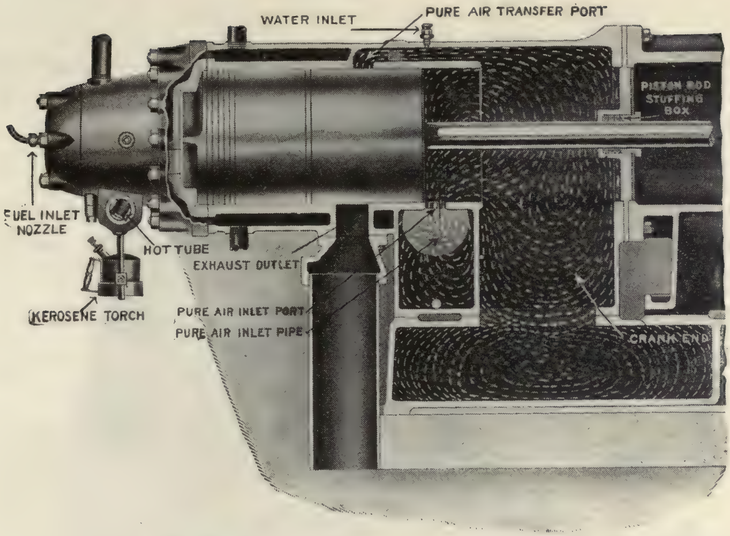
When a crosshead is not used, the piston must act as a crosshead and the cylinder as a guide. The piston must be made longer than otherwise necessary, in order to have room for the piston pin and to prevent as much of the inevitable excessive wear on both piston and cylinder as possible. This wear is caused by the piston being forced hard against the top and bottom of the cylinder by the angular thrust of the connecting rod.

This uneven cylinder wear can never be entirely prevented without the use of a crosshead. As the wear increases, it permits the oil of heavy base to work back and under the piston rings, hardening there, and causing additional wear. The advantages do not stop here. The extra friction caused by lengthening the cylinder and piston is greater than the friction of a crosshead. Engines which do not have crossheads soon become very hard to start on account of the loss of compression due to worn piston and cylinder.

Crosshead construction adds great stability to a machine. It has been definitely established that the addition of this one feature doubles the working life of an engine.

Hot Liner vs. Hot Ball or Electric Ignition: Electric ignition has not been successfully applied to the firing of low grade fuels. Engines utilizing this system are suitable only for burning kerosene and the more volatile fuels.

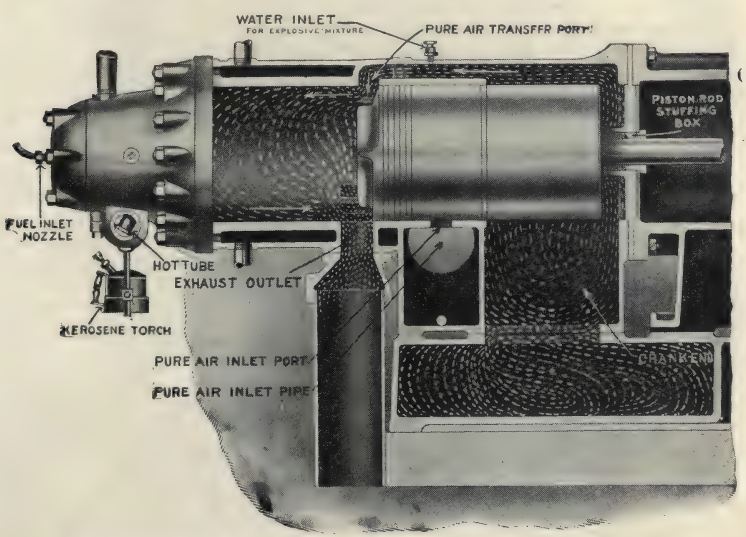
Ignition is secured in Giant Engines by injecting the fuel into a hot liner. This hot liner is not subjected to bursting pressures nor is it subject to breakage from contraction and expansion, as are hot balls sometimes used for ignition purposes in this type of engine. Hot balls collect



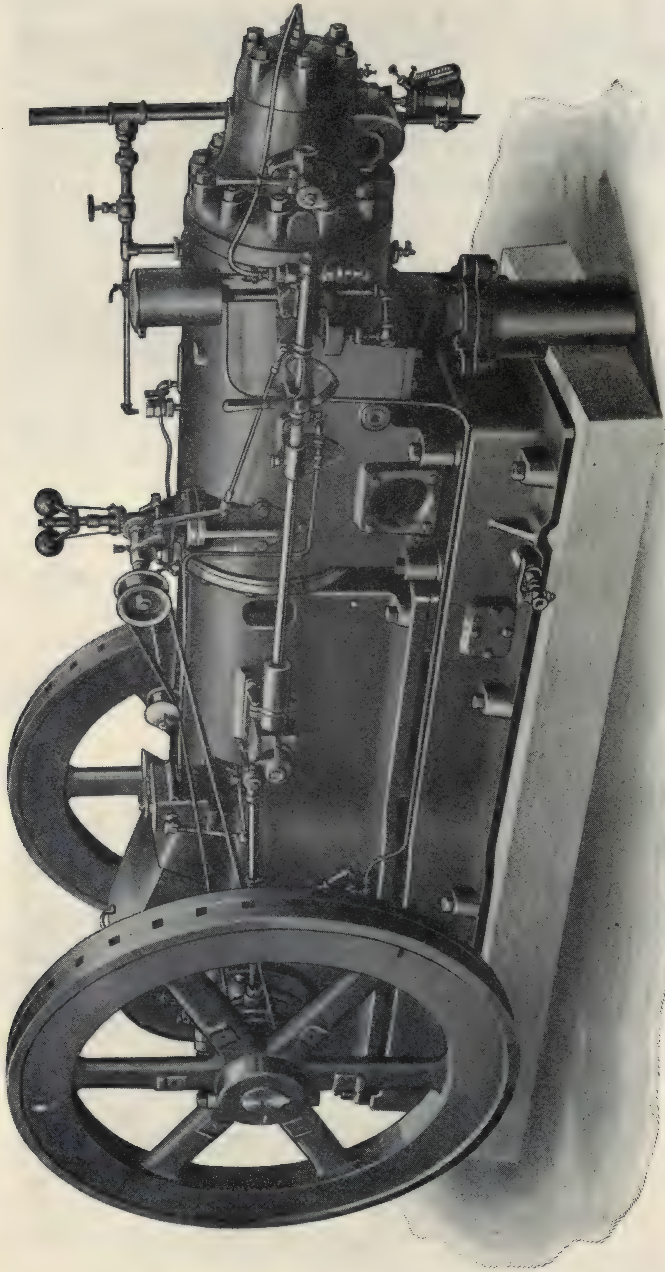
Position of Piston at Time of Combustion.

carbons. Especially is this true in engines in which water injection is not used.

In any engine using hot ball ignition the oil, upon its injection into the cylinder, comes in contact with very little heated iron as compared with the hot liner method used in Giant Engines. As a result it takes



Position of Piston at Time of Scavenging and Exhaust.



Giant Semi-Diesel Oil Engine (Governor Side), Class A-02.

much longer to gasify the oil which consequently must be injected into the cylinder much earlier in the stroke than when using the hot liner. The earlier the oil is injected into the cylinder of an oil engine, the more danger there is of pre-ignition and excessive initial pressure.

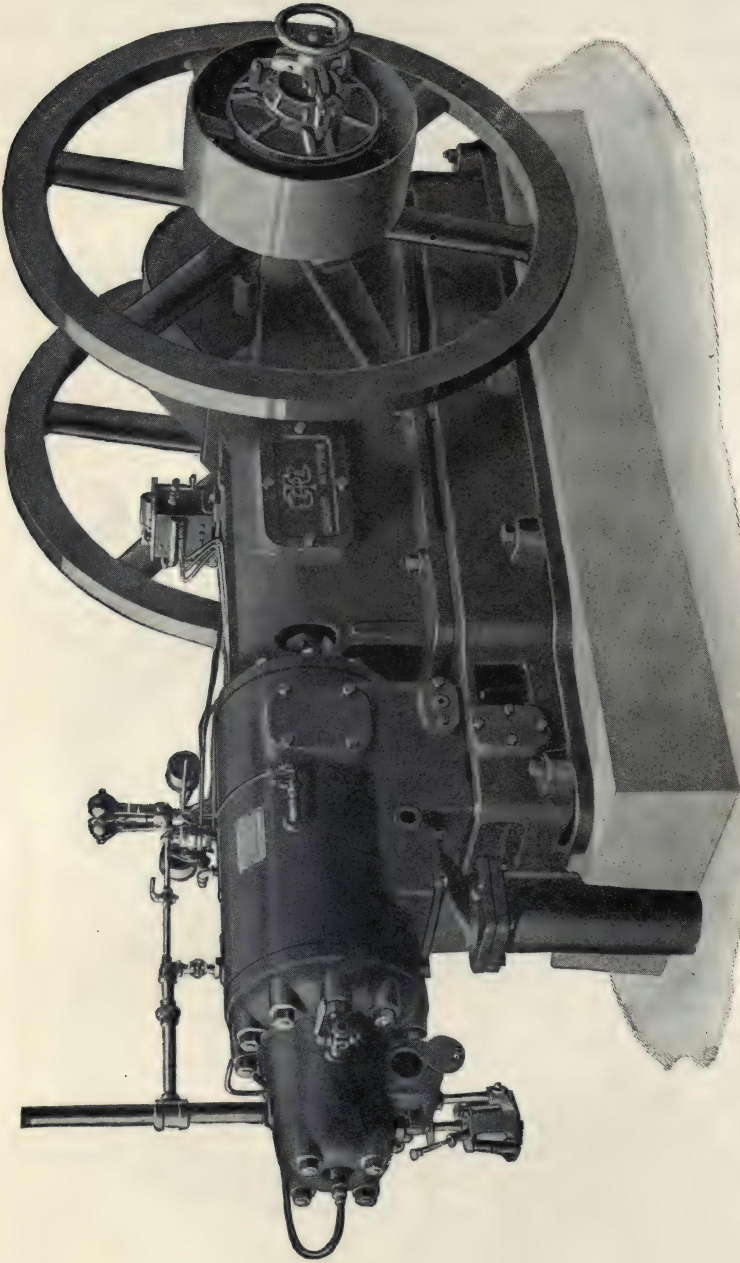
Further, any engine which relies for ignition on a bright red heat and which is subject to bursting pressure, is dangerous. In this connection the following table taken from the "Engineering" is interesting since it shows the decrease in tensile strength of cast iron and mild steel at various temperatures and was based on observations of hot bulb semi-Diesel Engine:

Load	Color	Temp. °F.	Tensile Strength, Cast Iron—Tons per Sq. In.	Tensile Strength, Mild Steel—Tons per Sq. In.
Light Load	Just showing color in the dark	750	12.0	24.0
Normal Load	Between dull and cherry red	1100	7.5	12.0
Over Load	Bright cherry	1400	3.5	2.5

The oil upon being forced into the combustion chamber in this engine, passes through, and strikes the head of the cylindrically shaped liner with which the combustion chamber is fitted. The shape of this liner is such that the oil is instantly distributed over its surface, gasified, and ignited. The resulting rapidity of ignition permits the injection of fuel into the cylinder late in the stroke, thereby avoiding the abnormal pressures incident to pre-ignition.

Giant engines are running on any petroleum distillate from 28 degrees Baume scale up to and including kerosene that does not contain any more than 1 per cent sulphur or 25 per cent asphalt. It is not recommended that any of the lighter distillates burned in gasoline engines be used. There are a number of oils considerably below 28 per cent Baume scale on which Giant engines will operate satisfactorily, but as this depends upon the character of the particular oil, a general guarantee cannot be given, as to the performance of satisfactory results, although recommendations to that effect are often given by operators. There are also many crude oils which can be used in these engines, but it is dangerous practice as they are likely to contain sand, grit or sulphur. The safest and most satisfactory oils are those furnished by the oil refineries. The average cost in operating a 50 H.P. engine should be from 20 to 25 cents per hour.

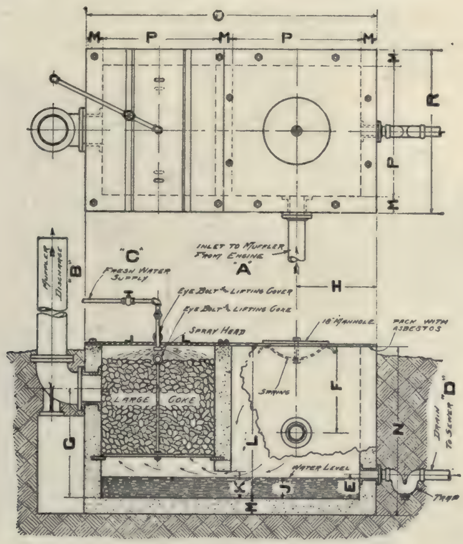
Water Injection With Fuel: Increased economy is secured by the use of water with the oil. The water enters the cylinders through a check valve at a point just above the pure air transfer port, and is drawn into the cylinder with the pure air. The water retards the combustion of the oil and thus keeps the initial pressure down to slightly more than compression pressure. It also keeps the cylinder free from carbon, keeps the piston rings from sticking, and aids lubrication, by helping to keep the piston and cylinder walls cool.



Giant Semi-Diesel Engine (Clutch Pulley Side), Class A-02.

Air Starter: On the Duplex Types of Giant Engines automatic air starters are provided. The compressor equipment for the system consists of a small vertical air-cooled single acting air compressor, driven by a gasoline engine of adequate horsepower. An air receiver of ample capacity, is provided, together with a pressure gauge, pop safety valve, and drain cock.

The automatic air starter consists of simple plunger valves bolted to the sub-base and operated by cams on the crankshaft. These valves allow a portion of the high pressure air to act on small piston valves,



—NOTE—
TO BE CONSTRUCTED OF REINFORCED CONCRETE

SIZE MACH H.P.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	R
20.	6"	10"	3/4	3"	7½"	2'-0"	3'-1"	1'-8"	6"	8"	4'-1"	8"	4'-9"	6'-0"	2'-0"	3'-4"
30.-50.	6"	10"	3/4	3"	7½"	2'-6"	3'-4"	2'-2"	6"	8"	4'-4"	8"	5'-0"	8'-0"	3'-0"	4'-4"
60-80.	8"	12"	3/4	3"	9½"	3'-0"	4'-0"	2'-10"	8"	10"	5'-2"	10"	6'-0"	10'-6"	4'-0	5'-8"

Muffler Pit for Single Engine.

one of which is attached to each cylinder head. These in turn admit high pressure air to the engine cylinders. The entire arrangement is remarkably simple.

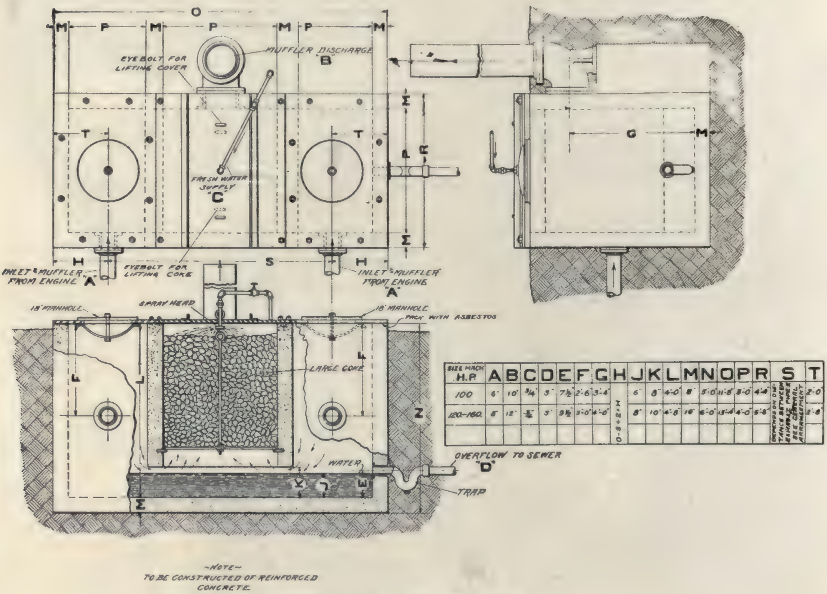
Exhaust Piping: Some suggestion is given here in regards to properly fitting of exhaust piping from the engine. In illustrations (f) and (g) a typical plan of installation of muffler pit is seen. When following the general instruction it will be noticed that particular attention is given to eliminate all undesirable noises caused by engine exhaust.

Bolt the exhaust flange to the under side of the cylinder and run the exhaust pipe from it to any point desired, taking care to see that the pipe does not come in contact with anything inflammable. If it is very

long or crooked, it should be increased in size. It should be put together in such a way that it can easily be taken apart for cleaning.

It is recommended that where practicable, the water jacket outlet pipe be connected to the exhaust. The introduction of the cooling water reduces the temperature, and deadens the noise of the exhaust. If this connection is made to an exhaust line in which a muffler is used, the drain at the bottom of the muffler **must** be left open.

When an exhaust pot is used, it should be placed as close to the engine as possible, and must be connected by the size of pipe called for by the openings in the exhaust pot and engine. When the line connecting the exhaust pot and exhaust outlet is long or if very many bends are made, it is recommended to use a size larger than that called for by the exhaust pot opening.



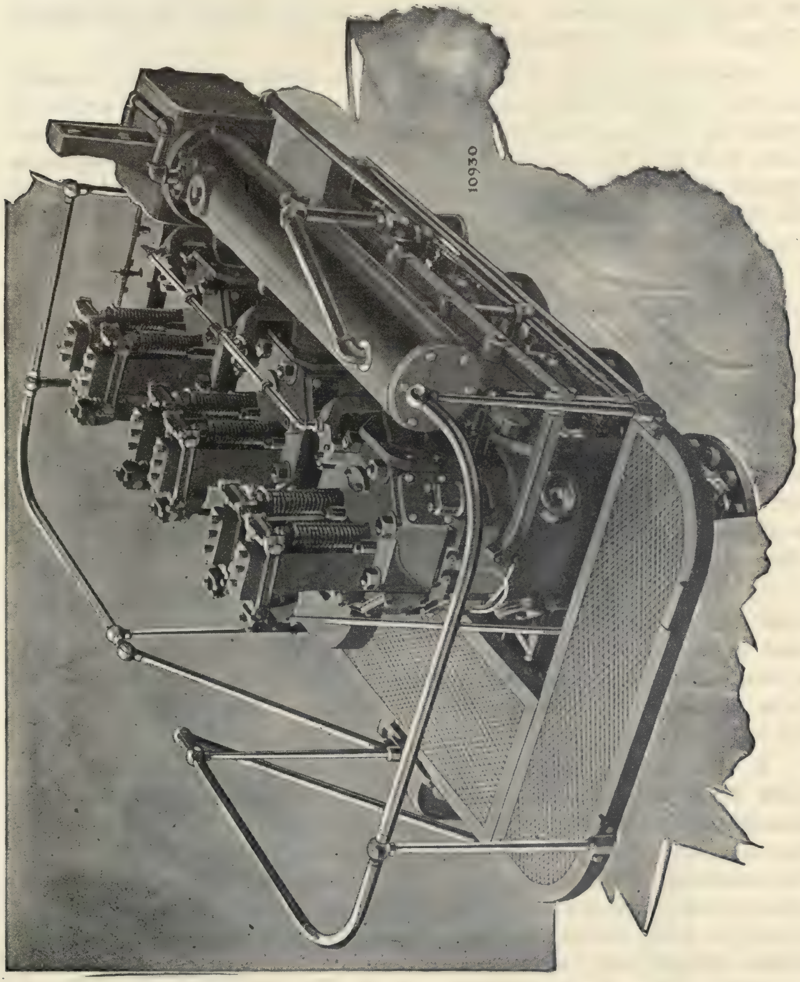
SIZE MACH H.P.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	R	S	T
100	6"	10"	3/4"	3"	7½"	2'-6"	3'-4"	O-S÷2=H	6"	8"	4'-0"	8"	5'-0"	11'-8"	3'-0"	4'-4"	DEPENDS ON DISTANCE BETWEEN EXHAUST PIPES. SEE GENERAL ARRANGEMENT	2'-0"
120-160	8"	12"	1/4"	3"	9½"	3'-0"	4'-0"		8"	10"	4'-8"	16"	6'-0"	13'-4"	4'-0"	5'-8"		2'-8"

Muffler Pit for Double Engine.

For installations where it is necessary that noise and smoke be eliminated, it is recommended that a muffler pit of the type shown in Figs. (f) and (g) be used. Figure (f) shows the design of muffler pits capable of taking care of single engines up to 80 H.P., while the design given in Fig. (g) will take care of Duplex engines ranging from 100 H.P. to 160 H.P. When a muffler pit of the type illustrated is used, it is absolutely necessary that an overflow of the size recommended be used, and also that a sufficient quantity of running water be used to carry off any residue or waste matter that may come from the exhaust. With the use of one of these muffler pits, only a barely perceptible puff of light smoke will issue from the discharge pipe.

INGERSOLL-RAND OIL ENGINES

Oil engines have been classified in a number of ways; yet those built prior to the present time can be divided into two general classes: the so-called Semi-Diesel, consuming 0.7 pounds of fuel per brake horsepower, which employs a hot bulb, hot cap or other hot surface to vaporize and assist in igniting the fuel; and the full Diesel type, consuming slightly over half the fuel, which employs a full water-cooled head, but high cylinder compression, still more highly compressed air to atomize and inject the fuel and mechanically operated spray valves to accurately control the fuel injection.



Air Plan View of Ingersoll-Rand "P-R" Oil Engine.

In the accompanying illustration we see the Ingersoll-Rand Oil Engine, which as a matter of fact falls in neither classification. It has just as high an over-all economy and is as fully water-cooled as the Diesel type, yet it demands no higher compression and no more complicated fuel injection system than the Semi-Diesel type. By an ingenious method of direct fuel injection, so perfect is the fuel atomization that 200 lbs. per square inch is quite sufficient to automatically ignite the fuel. It is not a surface-ignition engine. The Ingersoll-Rand oil engine is a distinct type; it is a low compression engine, direct injection, automatic ignition engine.

The engine uses the four-stroke cycle with low compression (about 200 lbs. per square inch), direct injection of fuel and has no other means of ignition than the temperature of compression.

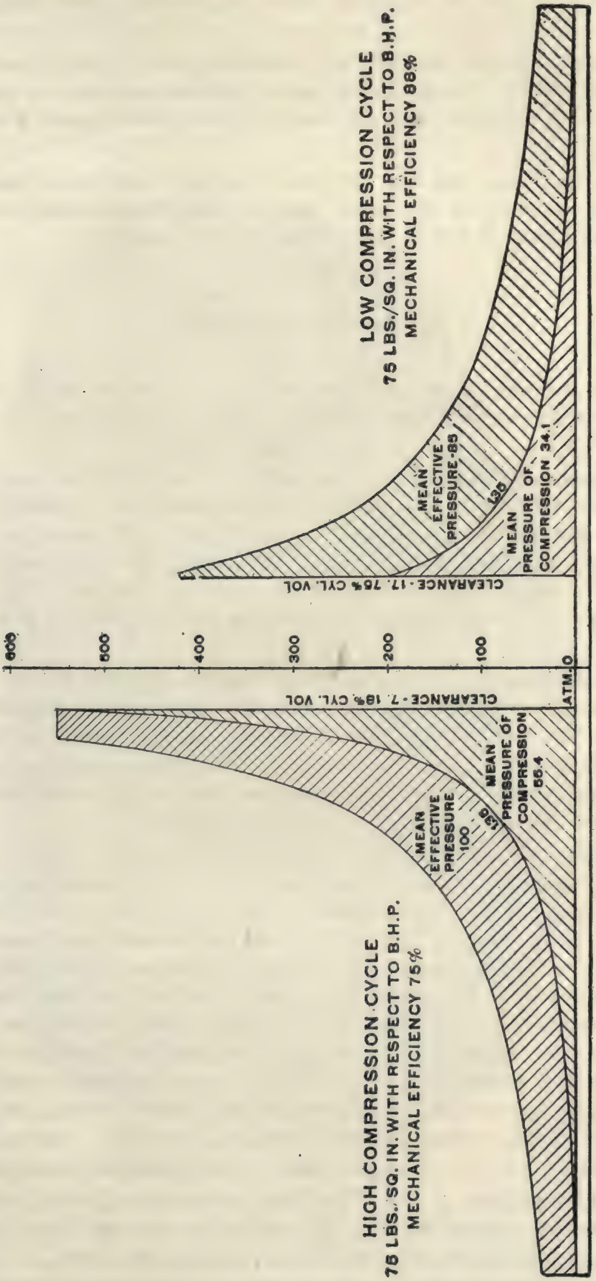
The advantages of solid injection are obvious, when we consider the two or three stage compressor necessary with the Diesel engine to inject the fuel into the cylinder under a pressure of necessary requirement including mechanical equipment necessary on high compression engines, whereas in this low compression engine, two fifths of that of the Diesel engine the same results are accomplished, eliminating all complicated mechanical contrivances.

In glancing over the efficiency card some idea as to the resultant economy of this engine will be demonstrated. It will be observed here, that after a pressure of 200 pounds to the square inch has been reached, combustion occurs at constant volume, creating a pressure of about 400 pounds, from which the expansion is almost a perfect adiabatic. Compare this with the high compression cycle, also shown in this chart, where the compression is carried to more than 500 pounds and the fuel is then admitted gradually so as to produce combustion at constant pressure until the piston has traveled a portion of the stroke, when it also changes to adiabatic expansion. There are three points of advantage in the low-compression cycle which should be noted:

1. The mean effective pressure which is proportional to the net work developed in the cylinder for the same brake horsepower of engine, need to be only 85 per cent of that of the high compression cycle. This is due to the higher mechanical efficiency. There are no air compressors to be driven with the Ingersoll-Rand Oil Engine, and the friction losses are in consequence lower.

2. The mean pressure of compression, which is proportional to the work done in the engine cylinder during the compression stroke, is approximately half that of the high-compression cycle. While it is true that most of this work is not lost, but being returned to the piston during its expansion stroke, nevertheless the performing of the extra work of compression by the piston and the return to the piston of an equal amount of excess work during the expansion stroke, represents more wear on piston, cylinder and bearings.

For the same brake horsepower the low compression cycle subjects the engine to approximately 30 per cent less wear. With parts of equal



Efficiency Card of Ingersoll-Rand Oil Engine.

dimensions they will wear considerably longer—they will require less attention.

3. The maximum pressure and temperatures in the cylinder are considerably lower, so that all the parts that are designed for strength, stiffness or temperature stresses may be constructed much more conservatively.

As the cycle process has features different from the average low compression as well as the usual type of high compression engine, it will be interesting to follow the accurate performance:

Description of the Cycle

Suction Stroke: The intake valve is opened mechanically and the piston moves downward on the suction stroke, drawing in a full charge of pure air.

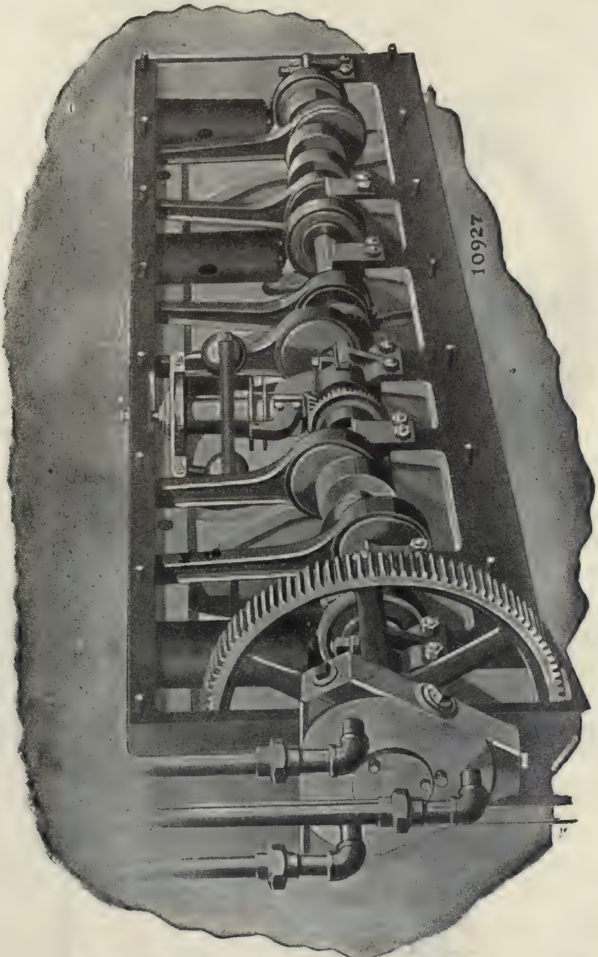
Compression Stroke: The intake valve is closed by the valve spring and the piston returns, compressing the air from the cylinder into the combustion chamber to a pressure of approximately 200 pounds per square inch. Injection of the fuel starts near the end of the stroke and is completed before the piston has reached the end of its travel. The system of fuel injection is such that ignition is automatic and perfect combustion occurs.

Working Stroke: Combustion at constant volume occurs almost exactly at dead center and the pressure rises from 200 to approximately 400 pounds and the piston moves downwards on the working stroke.

Exhaust Stroke: Near the end of the working stroke, the exhaust valve is opened mechanically, the pressure drops, and the piston returns expelling the burned charge.

The statement is often made that since the fuel cost of an oil engine is so low, a gain in economy is a small factor, that dependability is of prime importance. We fail to realize, that dependability is intimately related to high economy. Every heat unit in the fuel that is not transformed into useful work must be carried away through the walls of the combustion chamber or through the cylinder walls to the water jacket or must be carried past the exhaust valve and exhaust valve seat during exhaust, or must disappear as friction in bearings or cylinder at expense and upkeep and durability of the engine. In an inefficient engine, not only is a large percentage of the fuel wasted, but in getting rid of the waste heat, serious deterioration of the engine results. The waste oil must be paid for in the fuel bill, paid for in additional lubrication oil to preserve the oil film on the overheated piston and cylinder walls, paid for in additional cooling water to dissipate the waste heat, and paid for in engine upkeep. High fuel economy means low fuel cost, low lubrication cost, low cost of cooling water, low cost of repairs and long life of engine.

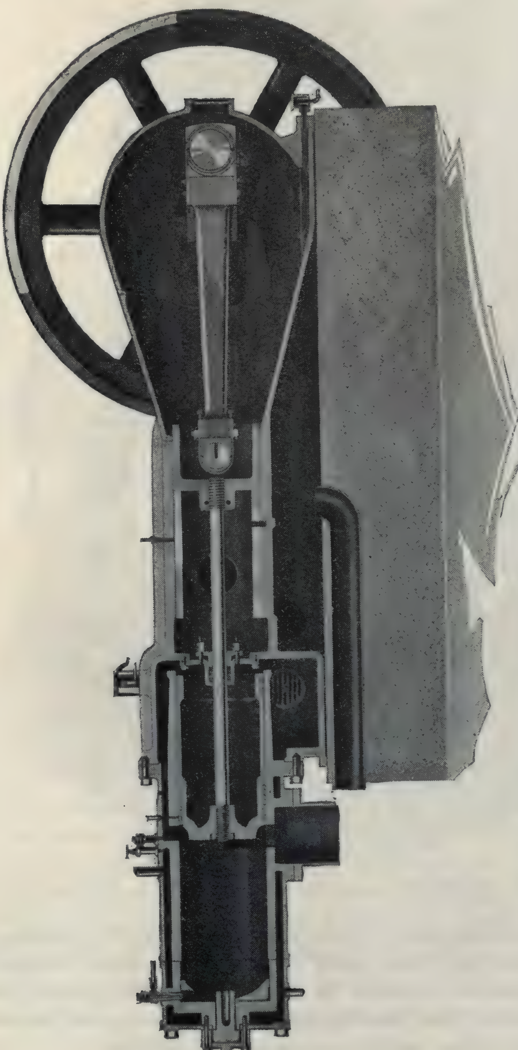
Valves on this engine are of the mechanically operated poppet type and located in the heads and surrounded by water jacketing. The valve motion is of the roller path type, operated by eccentrics mounted on the side shafts. This makes for quietness and smooth operation. The timing of the valves and the injection of the fuel is obtained from the side shaft through the medium of one pair of spur gears, driven by the crankshaft.



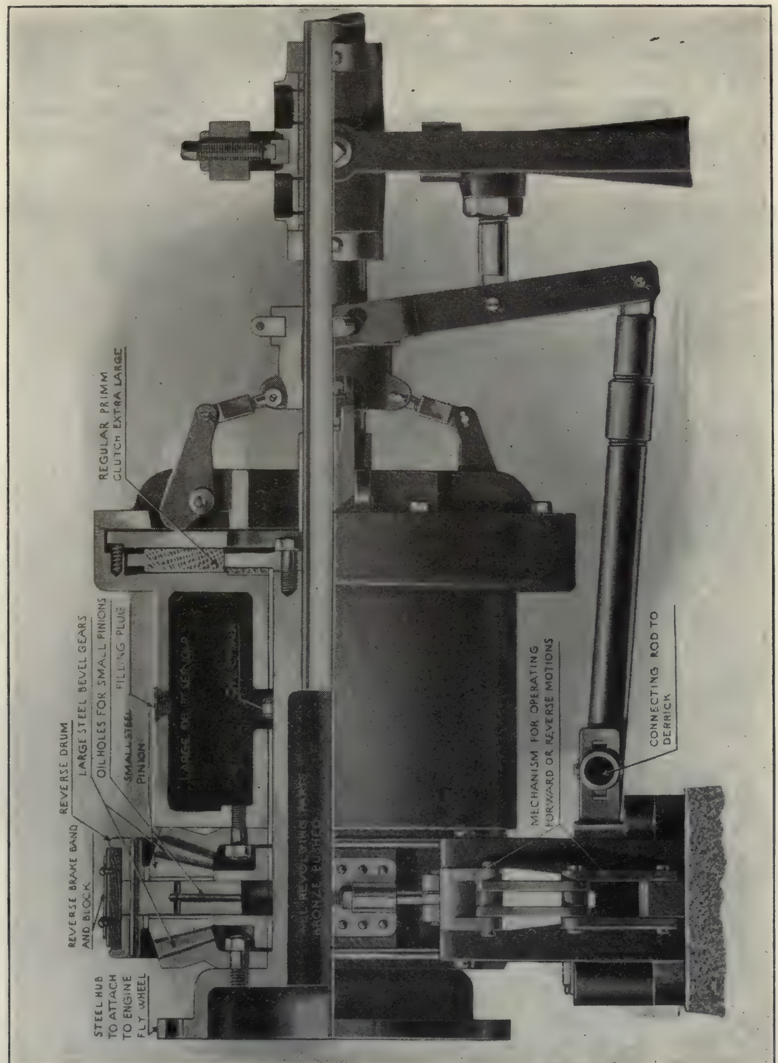
Front View of Engine with Cover Removed.

The roller path motion mentioned, consists of a floating lever, one end of which rests on the valve stem and the other is attached to the eccentric rod, which receives its motion from the eccentric on the side shaft. The end of the floating lever therefore moves up and down in relative motion. The upper surface of the floating lever is curved and

rests against a stationary block of slightly smaller radius of curvature. The point of contact between the two pieces changes as the valve opens and shuts; having the effect of uniformly accelerating the motion of the valve. It will be noted that at the moment of opening of the valve, considerable leverage is obtained on account of the point of contact being so close to the valve stem. This means that a minimum of stress is exerted on the push rods and side shaft when opening the valves against the terminal pressure. When the valve is opened slightly this pressure is destroyed, and the point of contact then recedes, so as to increase the speed of opening of the valve.



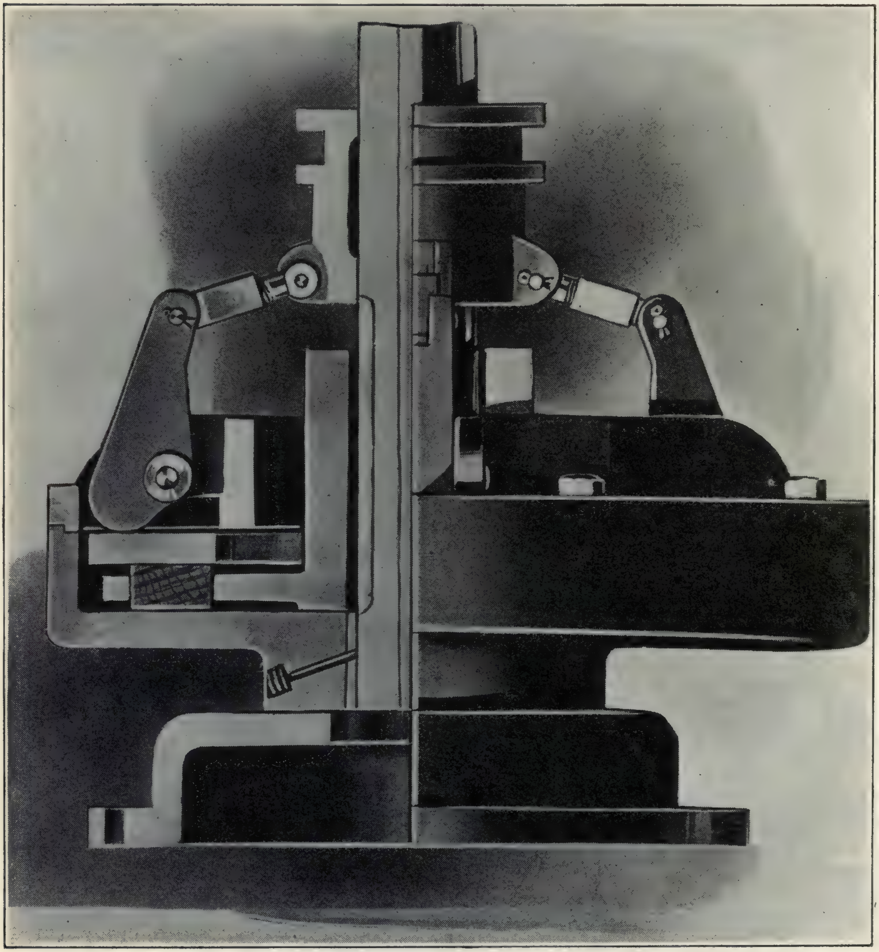
Longitudinal View of Single Cylinder Primm Heavy Duty Oil Engine.



Exposed View of Primm Reverse Friction Clutch.

The fuel injection pumps, one for each cylinder, which spray the fuel into the combustion chamber, are mounted on the housing adjacent to the cylinders. They are operated by cams from the side shaft. A centrifugal two-ball governor driven off the side shaft takes control of the fuel supply when the engine exceeds a pre-determined speed. It is an over-speed governor. An oil filter is provided from which the oil flows by gravity to the oil pumps. A pump, driven from the engine, elevates the oil to the filter from the main supply.

In starting, electric igniters are employed for the first few revolutions of the engine, while compressed air under 150 to 200 pounds pressure is admitted to each cylinder in succession by the aid of the starting valves, to turn the engine over. As the engine is turned over, fuel is injected and after a few revolutions the air is shut off and the engine continues to ignite entirely by compression. The engine may be stopped for about 10 minutes, and started again without the use of igniters.



Part View of Primm Friction Clutch Coupling.

In the accompanying illustrations the Primm Heavy Duty Oil Engine is shown. As will be observed this engine is of very simple design. It is of Semi-Diesel classification and is similar to the ordinary type in general method of running operation. The fuel consumption is no more than

six-tenths of a pound of fuel oil of a gravity of 24 Baume or better, containing at least nineteen thousand B.T.U.'s per pound, and containing not more than one-half of one per cent of moisture, the fluidity of which allow it to flow through the pipes leading to fuel pumps. This horsepower test and fuel consumption on three-fourths to full load is based on tests made at an altitude of approximately 1,000 feet above sea level.

It will be noted in the illustration of the engine, that the peculiarity of the ignition device is very distinct. Also the method of taking the air in the scavenging chamber, the cross-head taking all the angular thrust of the connecting rod; the enclosed crankcase and the splash lubricating system.

The use of the proper amount of water in the cylinder of an oil engine serves to maintain a proper interior temperature, upon which depends perfect combustion; this has been taken care of in this modern type of oil engine, by which the water injection into the scavenging charge by automatically measuring the amount of water needed with the same governor which controls the fuel injection, is adhered to.

In the illustrations pertaining to the Primm Clutch and Reversing gear, the ingenious method of providing the carrying of heavy loads by automatic governed mechanical gear arrangement, any load even under the most difficult conditions are amply taken care of and through this method no irregularities on the engine itself are experienced.

DE LA VERGNE OIL ENGINES

Medium Compression. Pump Injection System. Horizontal Construction

Of the numerous types of oil engines manufactured by the De La Vergne Machine Co., the two best known engines, Type "DH" and Type "SI" will be explained here.

Historical:

The Hornsby-Akroyd Oil Engine, better known as the De La Vergne Type "HA" Engine, was introduced by the De La Vergne Co. in 1893 for use in small power plants requiring up to 125 horsepower. The many admirable features of this type of prime mover soon won for it a position of great popularity. This engine employs a low compression ot about 50 pounds per square inch, and will operate on kerosene and light distillate oils. It has a comparatively high fuel consumption but great dependability. Many of these original engines are still operating after twenty-five years of service and examples may be found in various government lighthouses and fortifications.

To meet the demand for an engine of larger size which would successfully burn the heavy grades of American and Mexican crude oils, the Type "FH" Oil Engine was designed and offered in 1910, in sizes from 100 to 600 horsepower per unit. This type has a medium compression of 280 pounds per square inch and a compressed air fuel injec-

tion system. Its economy is greater than that of the Type "HA" and it operates on a wide range of oils, from heavy crude to kerosene. These engines quickly became popular for installations where the load

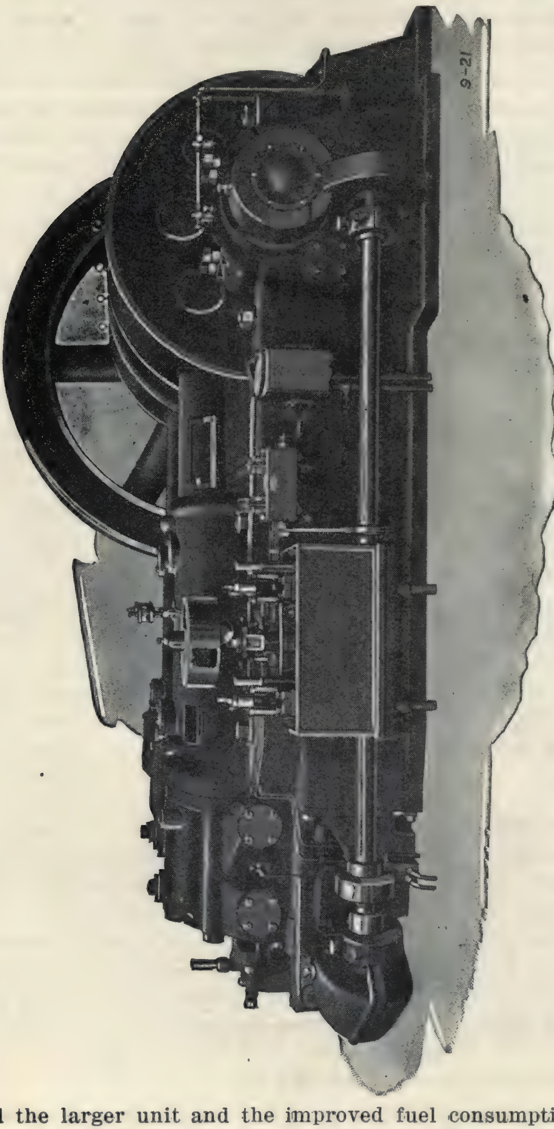


Fig. 1. 200 H. P. Integral Twin "SI" Engine

demanded the larger unit and the improved fuel consumption. One pipe line company operates nearly 150 De La Vergne Type "FH" Oil Engines.

In response to the demand for a small engine that, like the Type "FH", would be able to operate with high economy on the heavier and cheaper oils, the De La Vergne Machine Company developed and offered

the Type "DH" Oil Engine, in sizes from 40 to 130 horsepower per unit. These engines embodied the simple mechanism of the Type "HA" and some of the important features of the Type "FH" Engine. These Type "DH" Engines during the next few years showed such remarkable results in the way of dependability and low operating costs, that a demand naturally followed for an engine of this simplified design adapted to larger horsepowers. In response to this demand the De La Vergne Diesel Oil Engine (Type "SI") was brought out.

Cycle of Operation:

The Type "SI" Engine being a single acting four-stroke cycle engine, four strokes of the piston and two revolutions of the crankshaft are required to complete the cycle. The sequence of events follows:

Suction Stroke:

The intake valve is opened mechanically and piston is moved forward on suction stroke, drawing in a charge of pure air.

Compression Stroke:

Intake valve is closed. Returning piston compresses the air in combustion chamber to a pressure of approximately 330 lbs. per sq. inch. This compression pressure is ample to cause ignition because on account of the excellent atomization of fuel the entire combustion space is filled with a uniformly distributed oil mist.

Working Stroke:

Fuel is injected slightly in advance of inner dead center. Combustion occurs and pressure rises from compression pressure to a pressure of about 500 lbs. per sq. inch. Piston starts out on working stroke.

Exhaust Stroke:

The exhaust valve is mechanically opened, pressure drops and piston returns expelling the exhaust gases.

As previously stated, the Type "SI" engine is a single-acting, horizontal, four-stroke cycle engine, operating with a medium compression of about 330 pounds per square inch and a maximum pressure of about 500 pounds.

The engine is started automatically by admission of air from air storage tanks. To start, crankshaft is placed in starting position. Exhause valve roller under relief cam is shifted to reduce compression when starting. Starting air is then turned on. A cam on camshaft successively opens and closes starting valve, admitting air under pressure of about 150 lbs. at proper intervals. When engine picks up speed starting air is shut off.

To stop engine, simply pull out handle on fuel pump and lock it in position thus shutting off oil supply to cylinder and engine stops immediately.

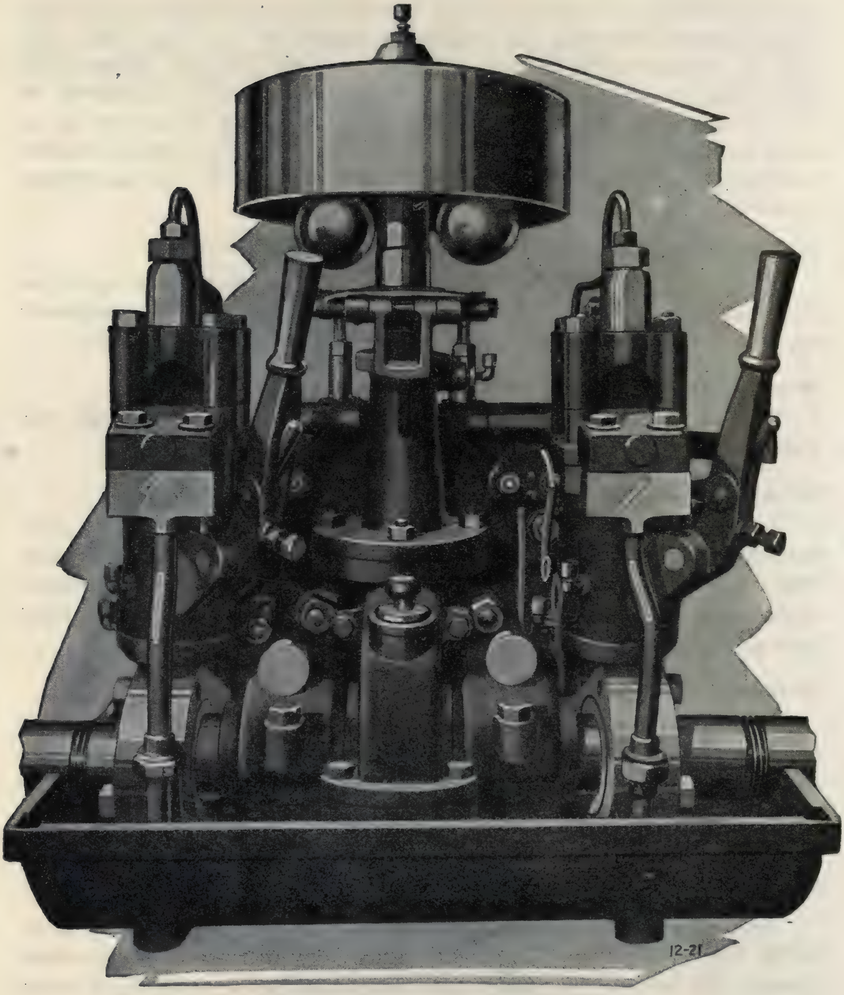


Fig. 2. Governor and Fuel Pump Arrangement

Figure 2 shows fuel pump and regulation. This is possibly the most important part of the "SI" engine. The pump is mounted on governor bracket and operated by a hardened cam on layshaft. Hardened steel pins and large working surfaces are used on all fuel pump parts. Pump plunger and plunger barrel are accurately fitted together and provided with labyrinth packing grooves, eliminating the necessity for the usual plunger packing. Suction and discharge valves with hardened and removable seats assure absolute tightness. Fuel is preferably stored in an underground tank outside the building from which point it is raised to a small filter standpipe by a plunger pump mounted on the engine.

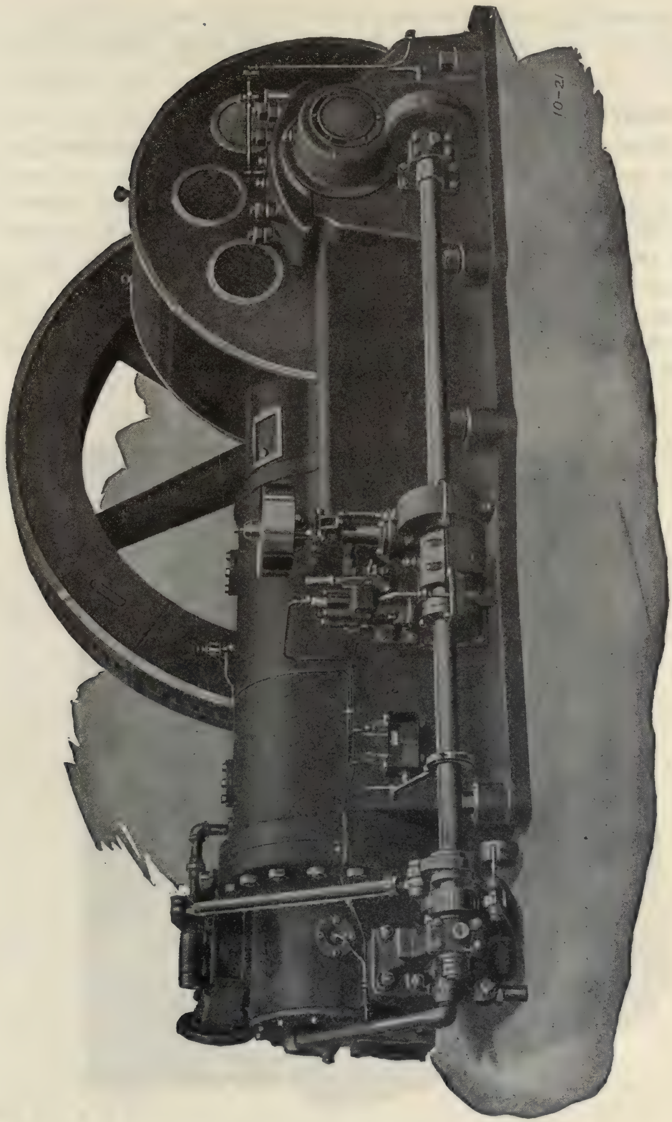


Fig. 3. 150 H. P. Single Cylinder "SI" Engine

When heavy viscous oils are used, standpipe is provided with a hot water jacket. Action of pump maintains constant flow of oil to standpipe, any excess fuel overflowing and returning to storage tank. From standpipe oil is withdrawn by engine fuel pump and delivered to spray valves.

The governor is of the centrifugal type operated by spiral gears from layshaft. The governor acts on an overflow valve on the pump

through a simple linkage so arranged that the quantity of fuel permitted to pass to the spray valve is readily controlled, thus regulating engine standpipe.

Cooling Water:

The Type "SI" Engine requires approximately seven gallons of cooling water per horsepower hour at full load. This water may be discharged at a temperature of 140° Fahrenheit. In locations where the water is costly or difficult to obtain a device of simple and inexpensive type will allow circulating water to be used again and again with the addition of only five per cent make-up water. Where water has objectionable scale forming properties a closed circulating system can be used which filled at the start with soft water, uses the same water repeatedly with practically no loss whatever.

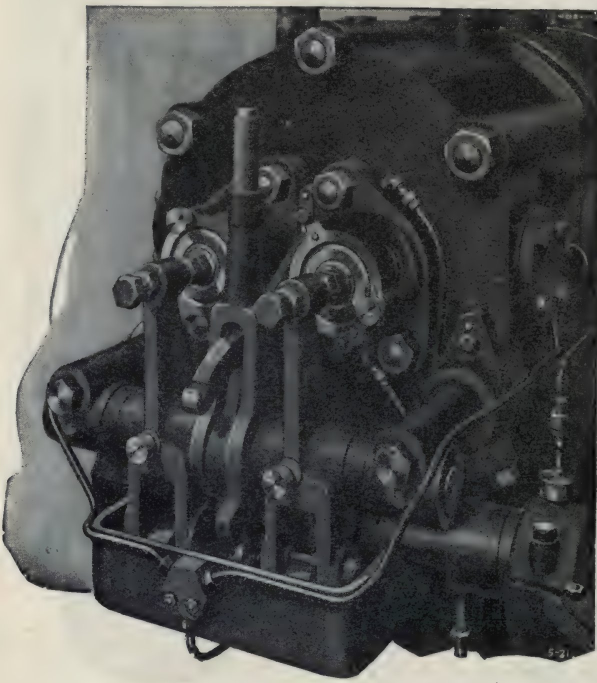


Fig. 4. Cylinder Head and Valve Gear

As will be observed in figure 4, the cylinder head is provided with an air starting arrangement. The camshaft side of the head is provided with openings for air starting valve and air and exhaust valve casings. The latter are made interchangeable and are mechanically operated from the camshaft.

There are two spray valves located on the opposite sides of the head connecting to combustion chamber which is cast on the inside of the cylinder face of the head. This arrangement insures complete combustion of fuel. The head is water cooled and provided with large hand holes for inspection and cleaning of water space.

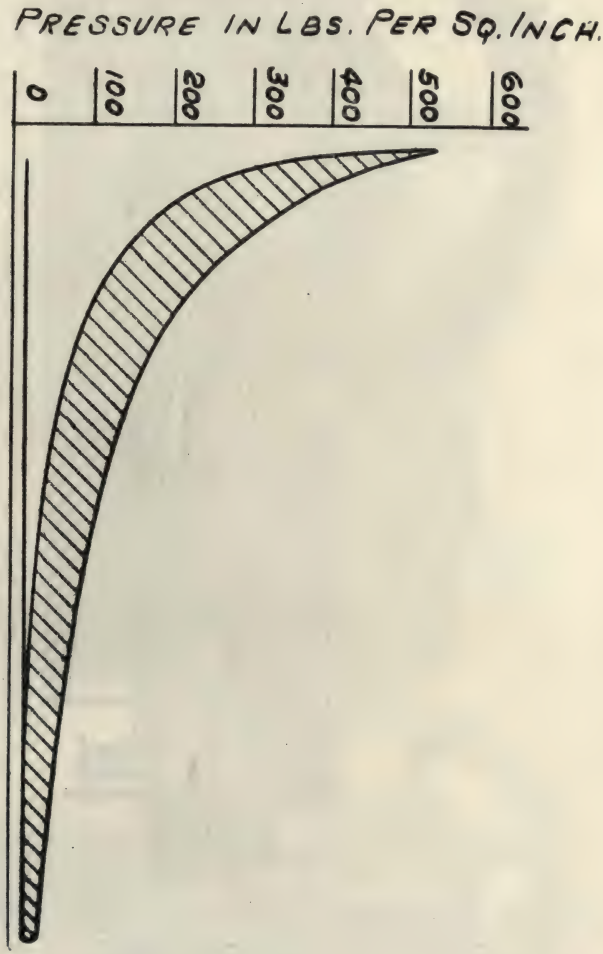


Fig. 5. Indicator Card

Piston:

The piston is of the trunk type and is made of a special heat resisting close grained gray iron. The casting is carefully annealed before machining and is afterward ground to exact diameter. The liberal length of piston enables the engine to reduce the pressure due to angular thrust of connecting rod to about 10 lbs. per sq. inch and pressure due to piston weight is less than one pound per sq. inch.

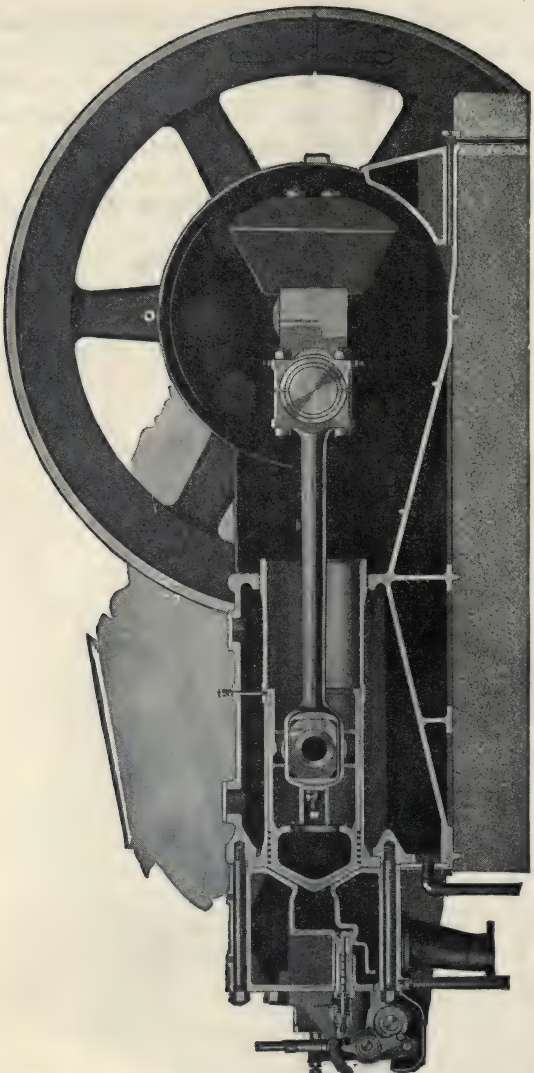


Fig. 6. Type "SI" Cross-Sectional View. Note Combustion Space

In this case the long trunk piston assures a large contact area between piston and cylinder liner, thus heat absorbed by the section of the piston head exposed to combustion chamber is rapidly conducted through piston and cylinder liner to cooling water. The piston therefore works under most favorable lubricating conditions, and water cooling of piston head with attendant complications is avoided. The highly polished head end of the piston is of cone shaped design, which is better able to follow expansion or contraction resulting from the working and exhaust strokes, thus relieving this important part of the engine of all internal stresses.

Rated Capacities:

The Type "SI" Engine is manufactured in the following standard sizes:

100 H.P. Single Cylinder	300 H.P. Twin Cylinder
150 H.P. Single Cylinder	360 H.P. Twin Cylinder
180 H.P. Single Cylinder	540 H.P. Three Cylinder
200 H.P. Integral Twin Cylinder	720 H.P. Four Cylinder

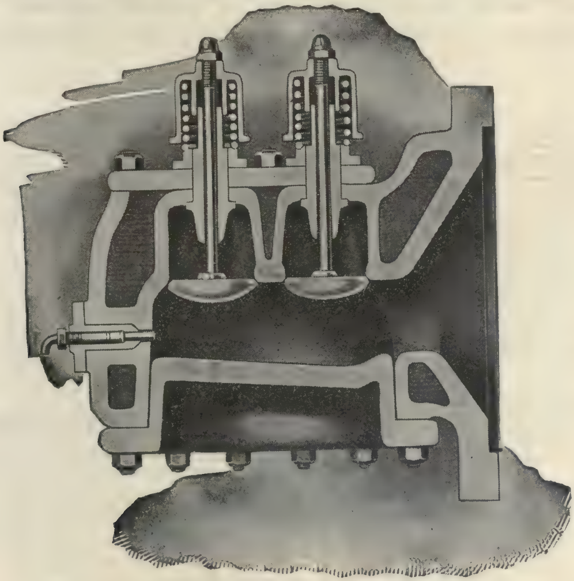


Fig. 7. Cross-Section Through Vaporizer

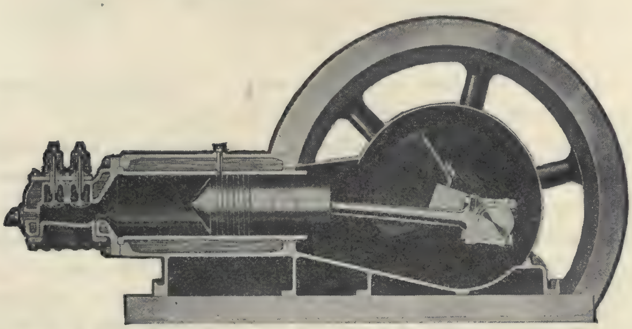


Fig. 8. Side Elevation in Section. Type "DH"

The reliability of an engine is merely a question of its weight and speed. As the weight is reduced, the reliability is correspondingly lowered. The Type "DH" engines weigh over 400 lbs. per horsepower, and are of massive construction compared to two-cycle units. Besides this,

the speeds of the "DH" engines are comparatively slow, which is conducive to longer life.

Operating at slower speeds and with all its working parts open to air, the "DH" engine is much cooler than many other high speed two-cycle units with its closed crankcase. The "DH" engine, introducing as it does a large volume of oxygen for combustion, injecting the fuel only at the end of compression stroke, employing a separate stroke for expelling the burnt charge, making use of exhaust induction to complete the scavenging and employing a separate stroke for drawing in the

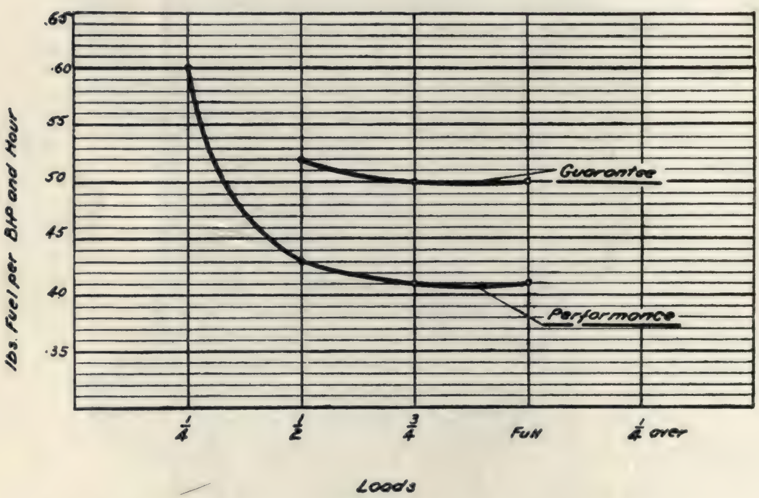


Fig. 9. Fuel Economy Curve of "DH" Type of De La Vergne

fresh charge of air, can be appreciated as a machine of high development among power producers of modern types.

The "DH" engines are equipped for automatic air starting. A suitable air receiver is provided. To start, the vaporizer is heated and the fuel pump is operated to inject the charge of oil into the vaporizer.

The air valve connecting to the tank is opened. This engages the automatic starting valve which is then successfully opened and closed by a cam on the layshaft. Compressed air from the starting tank is admitted to the cylinder and forces the piston outward. When the engine gathers speed the air is shut off and the engine then operates under its own power.

To completely burn heavy and tar constituent fuels and employ moderate compression the fuel must be thoroughly atomized and intimately mixed with the air in which it is to burn. The hot vaporizer walls facili-

tate ignition and subsequent combustion. in this way moderate pressures are employed, and the high efficiency of the Otto cycle attained.

The vaporizer, with its large heated area, makes nigh pressure in the combustion chamber unnecessary. The compression pressure need

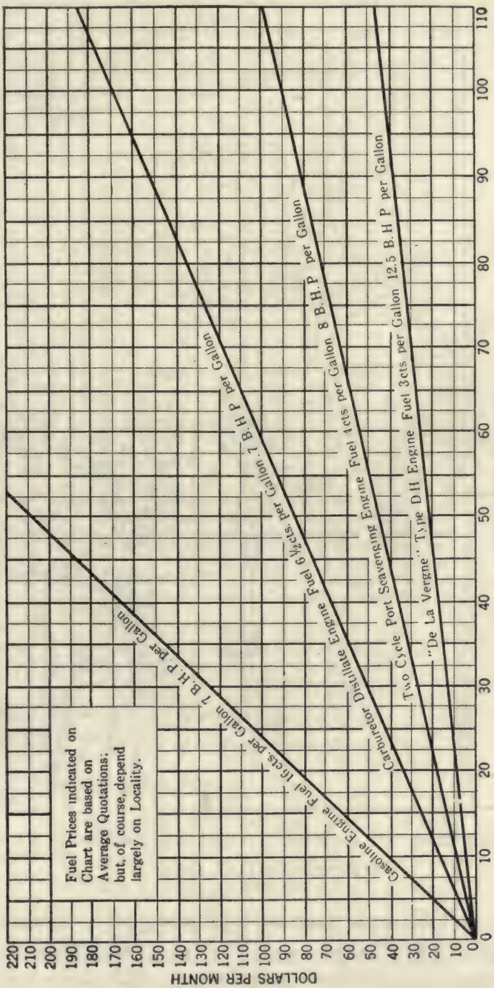
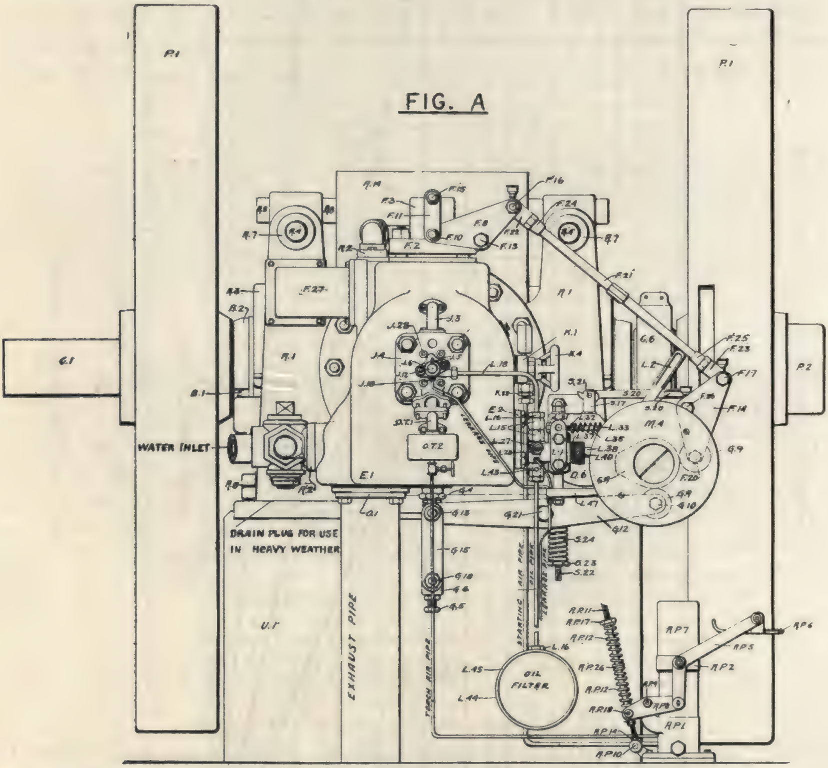


Fig. 10. Chart, Showing Comparative Fuel Cost of Various Engines

not exceed about 300 lbs. per square inch. Use of the uncooled vaporizer enables the De La Vergne engine to obtain as good results as those obtained with a pressure of only half as great as that used on high compression engines.

THE WYGODSKY SYSTEM OF OIL ENGINES

The Wygodsky Self-Starting Crude Oil Engine is manufactured by the Baltimore Oil Engine Company, Baltimore, Md., in several types, two of which are described hereinafter. One is a horizontal, four-cycle, heavy-duty, type for medium powers, and the other is a two-cycle type for large powers.

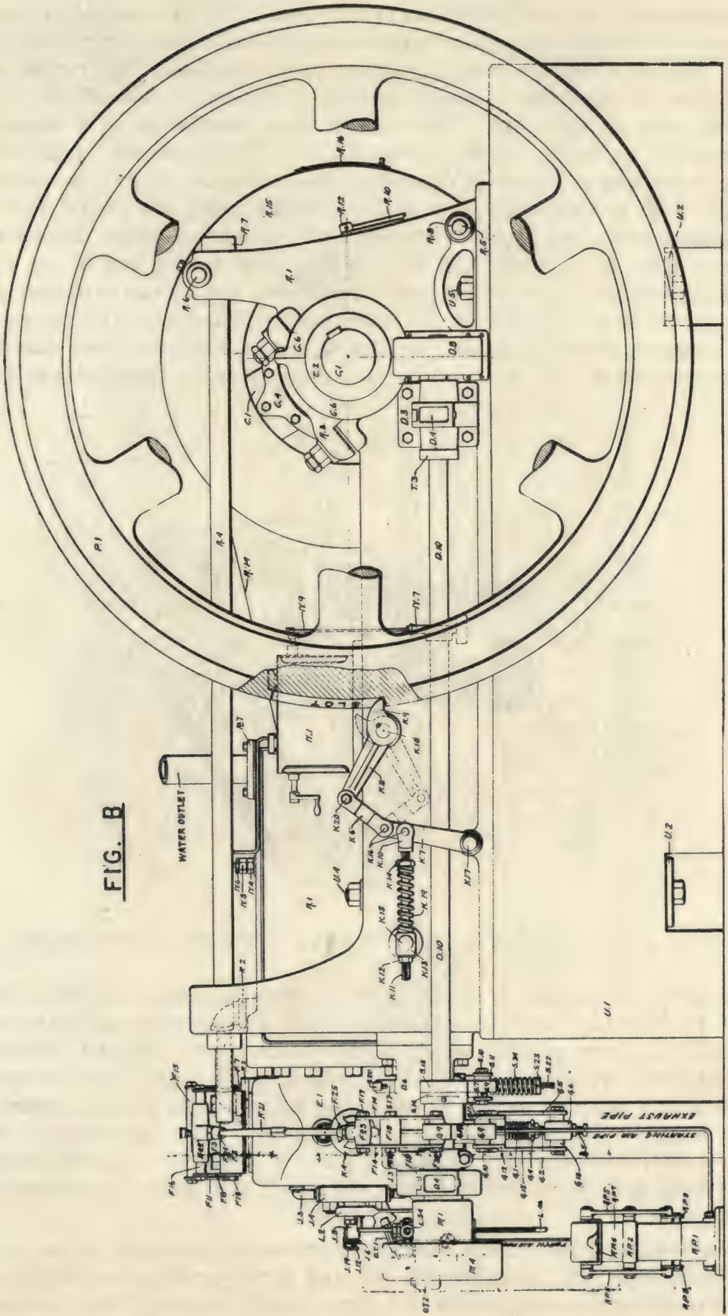


Wygodsky Self-Starting Oil Engine. Horizontal Type of Engine Showing Arrangement of Parts.

The horizontal four-cycle engine is illustrated in figures B and D.

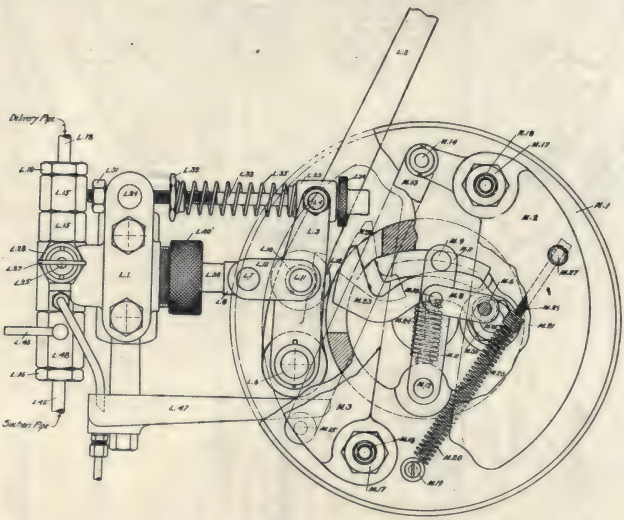
The principal feature of the engine is the airless atomizer, J.5, which in conjunction with the ignition element, J.1, the special fuel oil torch, OT, the air-foot-pump, AP, and the special flywheel locking device, K, makes the engine self-starting, without storing of compressed air, and without the use of electrical devices. These elements are described below.

The airless atomizer produces the atomization of the fuel by hydraulic pressure which is developed by means of fuel pump L (figure G.A.12) which is under direct action and control of the governor, M. The



Longitudinal View of Wygodsky Self-Starting Oil Engine.

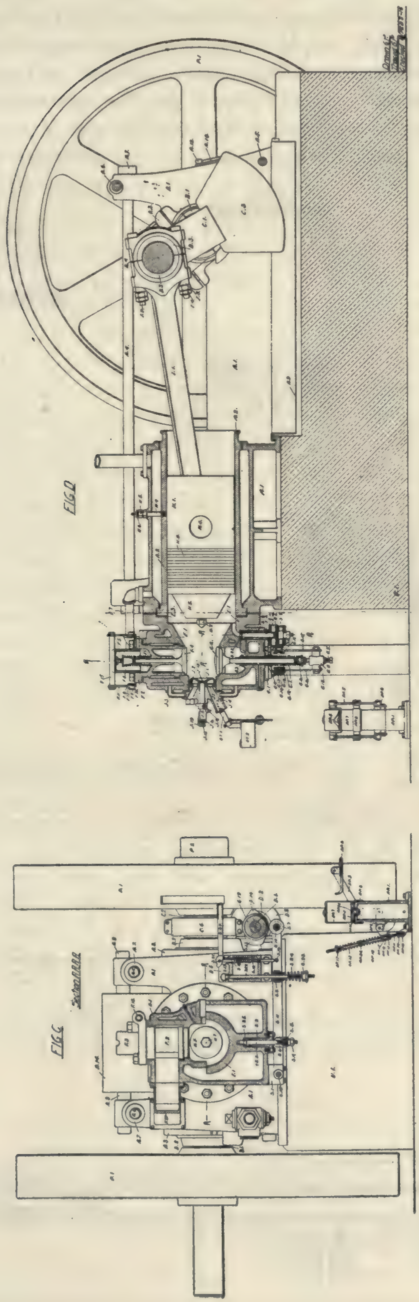
principal part of the atomizer is the spray cone, J.10, the conical surface of which is covered with spiral grooves of diminishing sections, and is placed against a similar conical surface in nozzle, J.15. The throat of this nozzle is closed by a needle valve, J.16, which is movable in conjunction with plunger, J.14. This plunger has a very snug fit in sleeve, J.18, and is loaded by means of spring, J.7. The pressure of the oil within the atomizer due to the action of the fuel pump, L, will move the plunger, J.14, against the spring and will carry along the needle valve, J.16, thus opening the nozzle. Oil will then circulate through the spiral grooves of the spray cone, J.10, with accelerating motion due to the decreasing sections of the spiral grooves, and will set up in the throat of the nozzle a small column of oil in high rotary motion. This column when released from the nozzle, breaks up into a very fine mist due to centrifugal action. As soon as the fuel pump plunger stops advancing,



Governor and Oil Pump of the Wygodsky Self-Starting Oil Engine.

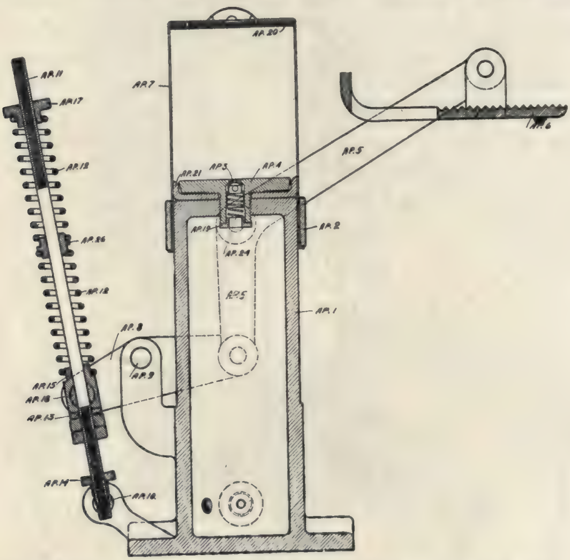
the pressure in the sprayer drops instantly and the spring, J.7, shuts the nozzle rapidly and prevents dribbling. This arrangement permits the breaking up of the smallest quantities of oil into very fine mist without any dribbling. To give any idea of the effectiveness of the spray, it may be mentioned that the column of oil in the throat of the nozzle produces about 1,100,000 R.P.M., and the initial velocity of the particles of oil which issue from the nozzle is about 270 feet per second. Any fuel, even with a high flash point, easily ignites when cold if atomized by means of this sprayer.

The engine works with a moderate compression pressure of 300 lbs. per square inch, and to obtain the starting ignition while the engine is cold, a temporary ignition device, J.1, figure G.A.10, is used. This ignition device is a hollow ring and is totally enclosed within the water cooled



Sectional Views of a Wygodsky Self-Starting Oil Engine.

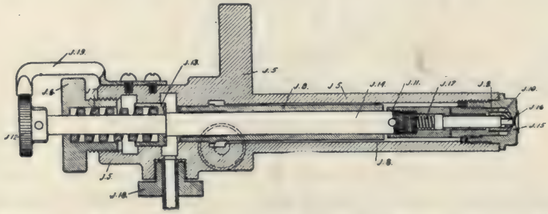
combustion chamber. No hot surfaces are exposed to the outside of the engine, but the heated surfaces are entirely inside the combustion chamber, so it is evident that this is not a hot bulb engine. This device has two apertures which connect the inside of this hollow ring with two channels in the water jacketed cover projecting through the cylinder head. One channel is for the introduction of the oil torch, G.A.13, and the other channel serves for the escape of the products of combustion.



General Arrangement of Air Pump of Wygodsky Self-Starting Oil Engine.

This torch, G.A. 13, connected to the foot-air-pump G.A. 15, which furnishes air to it, is of an interesting construction. The torch can burn the same fuel as the engine and the reservoir can be refilled while the torch is burning, as there is no pressure on the oil whatever. The torch does not require any pre-heating for it starts to burn as soon as the foot pump develops 25 lbs. pressure in a reservoir of about 3½ gallons capacity.

The next element of self-starting is the locking device, K, Figure B. This locking device consists of a toggle mechanism, K6, 7, 8 and 9. It is



Sectional View of Sprayer of the Wygodsky Self-Starting Oil Engine.

pivoted in two points, K17 and K18. It is loaded by means of spring K19. The position of this mechanism as shown in figure B, is that when the engine is ready for starting. The dog, K9, is then inserted in the slot of the flywheel. This keeps the flywheel in starting position, in which position the crankshaft is about 60 degrees above its inner dead center on the firing stroke. Sectional view, Fig. D, shows piston and crankshaft in starting position.

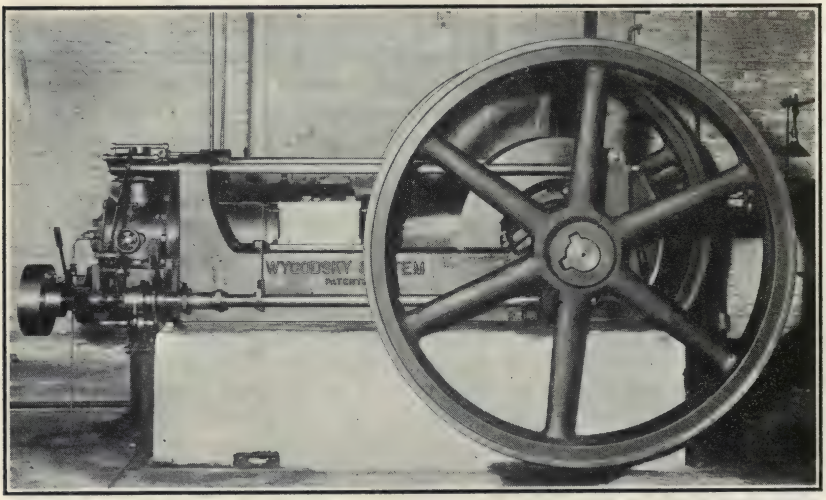
The process of self-starting is as follows:

The foot-pump, AP, is given several strokes and the torch, OT, is ignited by a lighted match, the pumping being continued for about two minutes, in the course of which time the ignition tube becomes hot enough to ignite the spray. After the ignition device has been made hot enough, some of the air from the foot-pump is admitted into the combustion chamber with the double purpose of making sure that the combustion chamber is filled with pure air and also to have the air pressure raised to about 40 lbs. The spring of the above mentioned locking device is made strong enough to hold the flywheel in the starting position so that the flywheel will not be released until there is enough pressure behind the piston to start the engine running. This pressure is produced by manually operating the fuel pump. This action produces a spray through the atomizer into the combustion chamber. This spray being produced in the proximity of the ignition tube and in an atmosphere of 40 lbs. pressure, an explosion follows which is strong enough to turn the dog of the locking device. This locking device then collapses as shown by the dotted lines in figure B, and the engine begins to work normally. The torch then goes out by itself, and the lubrication starts automatically so that no valves have to be operated. The starting cycle gives a M.E.P. above 100 pounds per square inch, so that the engine starts with a load of approximately 50 per cent of its rated capacity. The engine is also provided with special means for self-cleaning. By means of a special device the engine ejects automatically any solid or liquid deposit that may find its way into the combustion chamber. The same feature is utilized for stopping the engine approximately in its starting position. The M.E.P. obtainable in this engine is 116 pounds per square inch.

After the starting of the engine has been explained, the operation of same is easily understood by anyone familiar with a four-cycle oil engine. The governor which is explained below acts directly on the plunger of the fuel pump and injects the oil slightly before the end of the compression stroke.

The governor of an oil engine is one of the most important elements. Many a good oil engine proved to be a failure on account of poor governing. The method of governing in this Wygodsky oil engine is the variable stroke method; with timing of the beginning of injection, constant; and the end of injection, variable, i. e., sooner or later, according to the load on the engine. The governor of this engine comprises a governor proper, as well as the governing mechanism, all enclosed in one casing. The governor is also free of reaction from the governed mechanism and

therefore it permits running the governor at comparatively slow speeds, viz., the speed of the crankshaft. The governor and pump are represented in figure G.A.12, in which L.38 is the plunger, L.48 is the suction valve, and L.15 the delivery valves. The governing mechanism consists of the cam lever M.23, which is pivoted eccentrically in the governor casing. This lever is provided with a cam face which acts directly on roller, L.12, of the fuel pump. This cam face is an arc of a circle described from its pivot center. The position of this cam lever is determined by means of a finger, M.9, which finger is attached to a swinging lever, M.11, which is pivoted at M.12. This finger acts between a special inner face of the governor case and a curved face of the cam lever; the inner face of the case is an arc of a circle described from M.12 as a centre and the curve of the cam lever is figured out so, that the tangents of

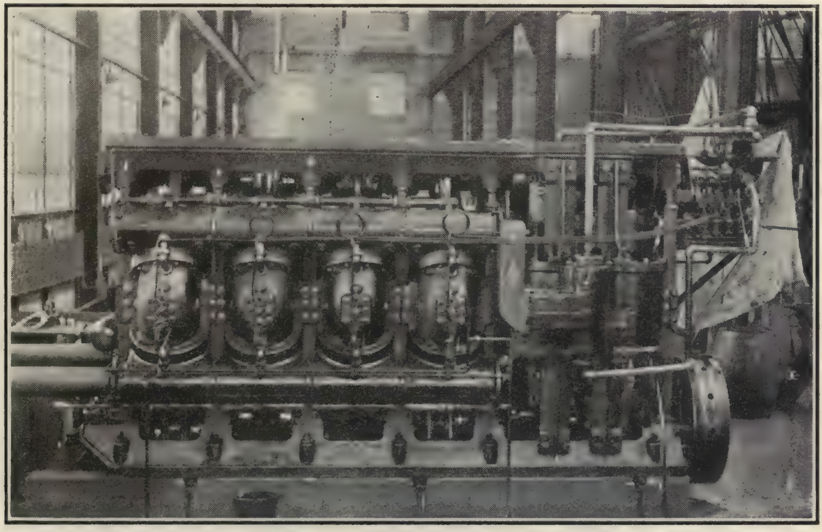


Four-Cycle Single Cylinder Heavy Duty Wygodsky Self-Starting Crude Oil Engine.

these two curves and finger, M.9, form an angle which is less than the angle of friction corresponding to the materials used. The idea is that when the governor runs in a counter-clockwise direction the reaction from roller L.12 on the cam lever, M.23, tends to jam the finger, M.9, without pushing same in either direction. It should be mentioned that the link M.11, by means of link, M.8, is connected to one of the governor weights, M.2. There is also another governor weight, M.3, which is situated in a diametrically opposite direction of M.2, and is connected to same by means of a connecting link, M.13, in such a manner as to give both weights a similar motion. These two weights furnish a centrifugal force of the governor when same is rotating. The centrifugal force is furnished by spring M.20, which is arranged to permit an adjustment of its length, as well as an adjustment of its tension. As mentioned before,

finger M.9 being situated in an angle which is less than the friction angle will not transmit any reaction to the weights of the governor. The cam face of the governor lever, being an arc of a circle, will have the same point of intersection with the outer periphery of the governor casing irrespective of the position of the cam lever. This naturally will give a constant point for the beginning of the injection of the fuel.

The whole engine is built on the interchangeable principle, and for easy operation, so that no expert labor is required. This feature makes it a suitable engine for any part of the country, as well as for export purposes.



Two-Cycle Wygodsky Self-Starting Crude Oil Engine. The economy performances compare well with the best.

Two Cycle Type

The principle of the Wygodsky Self-starting Crude Oil Engine is also utilized in a two cycle type engine.

This two cycle Diamond Type engine has been designed for engines of higher horsepower, and also for an engine which is much simpler in the design and manufacture than any other engine known.

The transfer of gases is accomplished through ports, which are controlled by the working pistons. It is a modified opposed piston engine, which, while preserving all the advantages of such a system, at the same time eliminates its disadvantages, such as multiplicity of crankpins, connecting rods, large dimensions, etc.

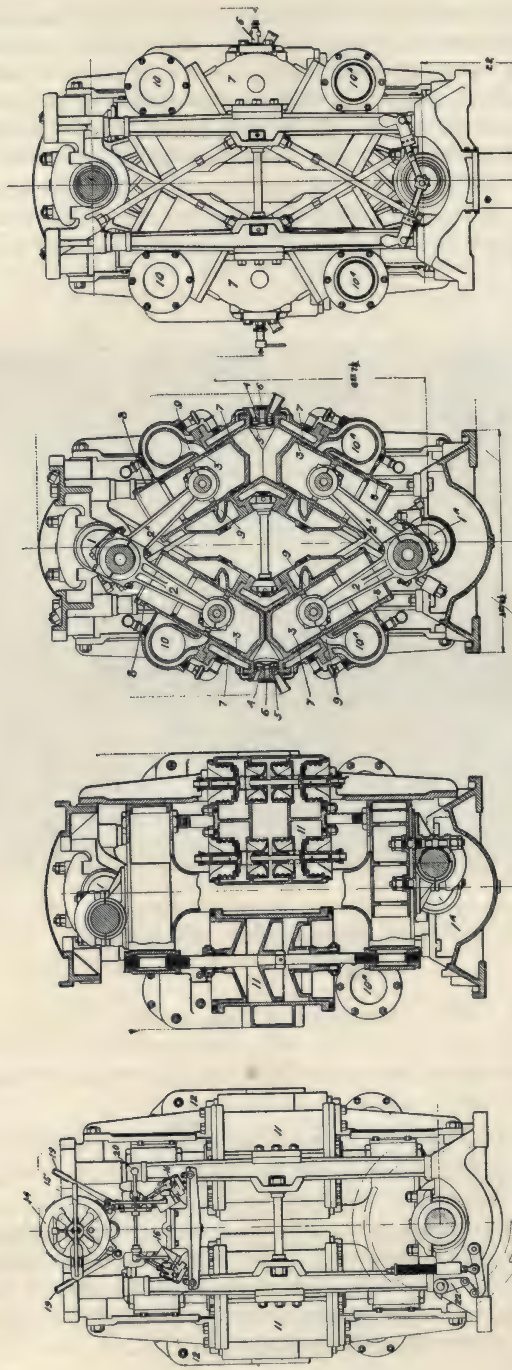


FIG. 1 FIG. 2. FIG. 3. FIG. 4.

Sectional and End Views of Wyyodsky Self-Starting Oil Engine.

This engine works with the previously described patented hydraulic sprayer, **without compressed air**, and has a low operating pressure. It is easily started from the cold in about three minutes, and uses very little compressed air, when starting with compressed air, and still less when starting with the self-starting mechanism, which has been described in connection with the stationary engine. The camshafts are eliminated, besides valves, levers, brackets, rollers, etc. The whole engine is just a mass of reciprocating pistons and revolving crankpins.

It has been recognized of late that an oil engine, to give a large output of power, must necessarily be of the two-cycle type. Although the opposed piston principle has been used many years ago in gas and gasoline engines, its adaptation for oil engines has been perfected only recently. This system primarily permits perfect scavenging, eliminates air and exhaust valves, as well as mechanism for operating same, and also eliminates the cylinder head, which is a very weak part in any oil engine.

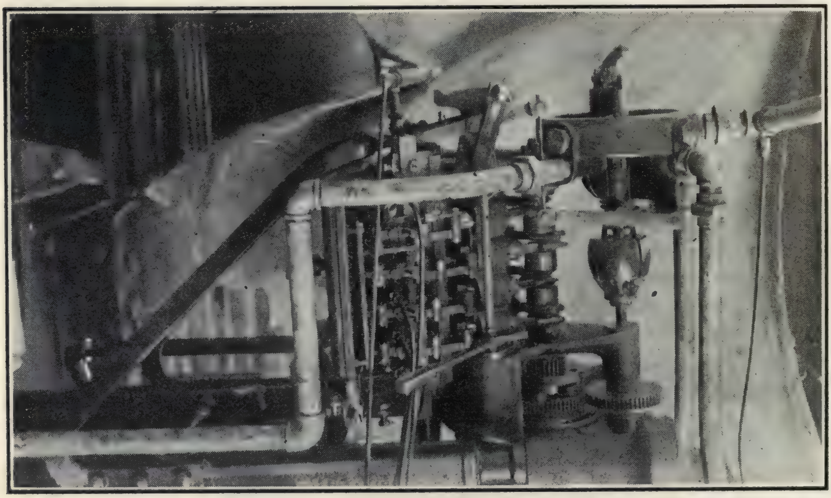
The two pistons, which operate in a common combustion chamber, permit the development of double the power with the same mechanism for the introduction of fuel. This has very many attractive features, although the way it has been carried out up to this time, has many great mechanical drawbacks. For instance, to operate the two pistons it is necessary to have three crankpins and three connecting rods, besides all the appurtenances that go with same. Moreover, it is practicable to put only two bearings between said crankpins, and therefore the crankshaft has to be made of very heavy sections, and naturally the crankshaft is expensive. Furthermore, the width and length of such an engine is very great, and therefore very costly to produce. It will be seen in the engine described below that only one crankpin is required for two pistons instead of three; furthermore, a group of four pistons requiring two crankpins are located in one plane and are operated by two crankpins located in two separate crankshafts. The regular opposed piston system requires 6 crank pins for four pistons. From the above the compactness and small dimensions of the Wygodsky engine is easily seen. This method readily permits the concentration of **large powers in a small space.**

Figure 3 shows a sectional view of the engine. As intimated before, this engine is provided with two crankshafts; one upper crankshaft, 1, and lower crankshaft, 1A. They are connected by means of a special link motion so that they operate in synchronism and in opposite sense. Each crankpin is operated by two connecting rods: one left-hand connecting rod, 2, which is forked, and the other right-hand, 2A, operates within the fork of the former. The pistons, 3, are operated by said connecting rods, so that each upper and lower piston work symmetrically. The cylinder structure is made in a V shape and this permits accommodating two such structures, one on each side of the crankshafts plane. These two cylinder structures give the section of the engine a "diamond shape."

In figure 3, four (4), indicates the spray box to which is attached the

starting ignition ring, 5, and the atomizer is introduced through aperture 6; both essentially the same as in the stationary four-cycle engine.

The cylinder proper, 7, is a water jacketed "elbow," the lower leg of which is slightly shorter than the upper leg. The reason for this will be explained later. This cylinder is designed to withstand the full pressure of the working cycle. To the upper, as well as the lower flanges of said cylinders, 7, are attached guide cylinders, 8, which receive the pistons after the ports have been uncovered. These guide cylinders are provided with port belts, 9, which extend all around the piston, and the contour of which is made to conform to that of the bottom of the piston. This belt, 9, is in communication with passages, 10, and 10A; the former serving as air manifolds, and the latter being exhaust manifolds. Thus we have two air manifolds, and two exhaust manifolds. These manifolds are provided with special flanges and when such cylinders are assembled



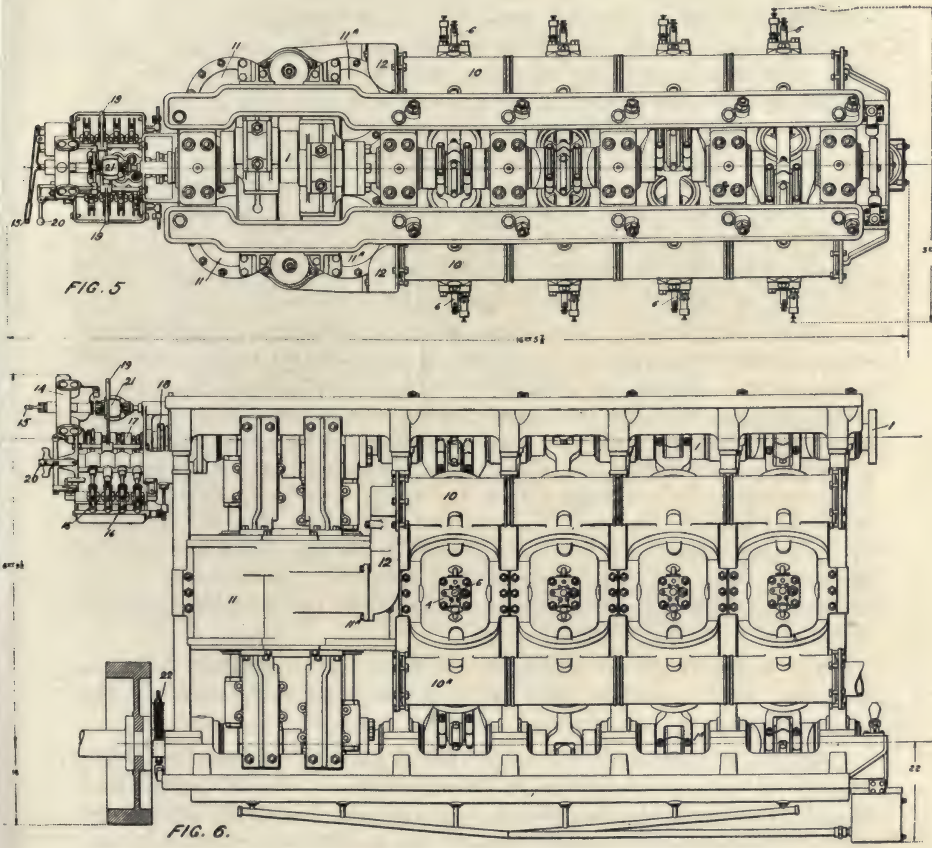
Fuel Pump Bracket of Two-Cycle Wygodsky Self-Starting Crude Oil Engine.

in the engine they register with each other and form continuous passages for either air or exhaust. They are kept air tight by means of a special packing device. Thus no special air or exhaust manifolds are required.

It will be noticed that the bottoms of the pistons are provided with flats which are mutually parallel in each cylinder structure and thus provide a combustion space which is much different than the wafer shape which we see in the regular opposed piston engines or even in the regular Diesel engines.

Figure 5 is a plan view, and figure 6 is a front elevation of the engine. On the left hand side of said figures, 11, and 11A indicate the two double

acting scavenging pumps, which by means of tubular member, 12, are connected to the two air manifolds, 10, into which these two pumps discharge the air. The four pistons of said pumps are operated by means of four crankpins, two of which are located in each of the two crankshafts respectively. The same crankpins are utilized for the link mechanism which connects both crankshafts so as to make them operate in opposite directions; thus the crankpins serve a double purpose, i. e., oper-



General Arrangement of Baltimore Oil Engine. Vertical Type.

ating the four pistons of the scavenging pumps as well as the connecting mechanism between the two crankshafts. Owing to automatic air valve of a special construction these same scavenging pumps also serve for starting and reversing the engine by means of compressed air. The different parts of the pump are connected by means of pipes to the air distributing mechanism, 14, which is actuated by means of hand lever,

15. This hand lever when pushed forward starts the engine in a forward direction; when pushed backward it starts the engine in a backward or reverse direction. When left alone the lever automatically returns to its neutral position and automatically cuts off the air supply from the tank.

The movement of said lever, 15, has the effect of automatically converting the scavenging pumps into working cylinders directing the air into the necessary chambers to effect the movement of the pistons in the desired direction and finally exhausting expanded air into any of the working cylinders, scavenging same through the usual ports. After that the air escapes through the regular exhaust pipe. This system of utilizing the compressed air has the advantage of securing pure air for the combustion and also does away with the necessity of providing extra mufflers to take care of the air exhaust. Needless to say that the complicated controls which are met with in the starting mechanism of the Diesel engine are completely done away with. It also must be mentioned that this engine besides ease of starting and reversing, uses very little air to effect these actions. It is necessary just to give about one-third of a revolution to start the engine spinning in any desired direction; this is due to the above described patented ignition device.

As mentioned before, the engine has no camshafts. The whole operating mechanism is assembled on one bracket, shown in the upper left hand corner of figure 6, in which 16 shows four fuel pumps; four similar pumps are on the other side. Each pump is connected to its respective combustion chamber. All fuel pumps are operated by a small crankshaft, 17, which is similar to the main upper crankshaft and is attached to same by means of a special clutch, 18, which permits a certain angular movement between the two crankshafts. The action of each pump may be tested by means of lever 19. Lever 20 is for regulating the speed of the engine, and when raised raises the speed and when lowered reduces the speed of the engine. Governor, 21, regulates the maximum speed of the engine.

For self-starting, the engine is provided with a locking device as described in connection with the stationary engine. Parts of this locking device, 22, are indicated on figures 1 and 6.

From the description above, it can be easily seen that the system permits a large concentration of power pistons in a small space, so it is not surprising that this engine is very light in weight (35 lbs. per B.H.P.), although the pistons speed is only about 750 feet per minute.

For operating the engine, two methods of starting will be described: one, the air starting, the other, "Self Starting."

It is recommended in regular practice, for starting as well as reversing, to use the air device, while the "self starting" is utilized in emergency cases when for some reason or other there is no air available.

Before starting, the ignition tubes, 6, are heated by means of special torches inserted in the lower funnel of spray box, 4. This heating is continued for about two minutes. After this, air-starting lever, 15, is

slightly moved either forward or backward, depending upon the direction in which it is desired to set the engine in motion. The compressed air will then operate the scavenging pumps and as soon as the engine has made about one-third of a revolution, fuel pumps will come into action and the engine will begin to run on its own power. As mentioned before, as soon as the hand is removed from the starting lever, 15, same will automatically return to its neutral position; this will cut off the compressed air supply and the pumps, 11 and 11A, will work as "scavenging" pumps.

For "self starting," the engine must be first set in a "starting position," i. e., a certain cylinder must have its pistons in such a position that its corresponding crank pins are about 60 degrees beyond its inner dead centre, as shown on the right hand side in figure 3. The slot in the flywheel, as well as the locking device, are so situated that this is attained in the first working cylinder on figure 6, counting from the left.

After the flywheel has thus been "locked," the torches are ignited and kept burning for about two minutes, as for air starting. Then air at about 80 lbs. pressure, is introduced into the combustion chamber between the two pistons. A quick push by hand of the corresponding oil pump plunger, will send a spray of oil into this combustion chamber, which will be readily ignited by the hot ignition tube. An explosion will follow, which will collapse the locking device holding the flywheel, and the engine will start operating. It must be observed that a very small quantity of air is required for this method of starting, and said air can be easily generated by several foot pumps, same as used with the stationary engine. Same pumps will also serve to operate the torches.

For reversing the engine, speed lever, 20, is first put in the bottom position and the starting lever 15 moved in the direction it is desired to run the engine. Just a slight movement of the crankshaft is sufficient to start the engine running. The engine requires very little air for reversing.

It is possible to reverse the engine without compressed air by means of a special device giving a premature ignition.

As soon as the engine is started the torches are extinguished and after that all operations such as running, maneuvering, and reversing are performed without the torches burning.

The method of operating the engine does not require much explanation. As seen in figure 3, the pistons approach each other in a symmetrical fashion and when about at the end of their inward stroke, a fuel pump injects the oil through the atomizer, which oil is ignited. On the expansion stroke, the lower ports, 9, are opened first and the combustion space is then in communication with the air manifold, 10, in which there is slightly compressed air. This air will then pour into the combustion space, driving before it the products of previous combustion and scavenging the said combustion space thoroughly, as there are no valves or other kind of pockets in the whole structure. The return

movement of the pistons will first close the transfer ports and then the exhaust ports, and then follows the compression, etc.

The advanced opening of the exhaust ports is accomplished by making the lower leg of the V cylinder structure slightly shorter than the upper one.

In the construction as shown, the ports are so arranged, that in any position of the crankshaft, there are always open some transfer and exhaust ports, and so the air manifold, 10, is always in communication with the exhaust manifold and the outer atmosphere through one of the cylinders and therefore there is very little pressure maintained in this manifold, just enough to overcome the frictional losses of this pipe system. This feature is of great importance as the scavenging pumps, although of about 50 per cent over-capacity, have to work under very little pressuré and while insuring perfect scavenging, the pumping losses are negligible as compared with most of the two cycle engines in which the air in the manifold is kept under several pounds pressure.

As constructed, the engine produces 8 double impulses per revolution. It has six upper crankpins and six lower crankpins—a total of 12 crankpins. In the well known opposed piston engine to obtain the same number of impulses, 26 crankpins would be required, this including 2 crankpins for operating two scavenging pumps.

A four cycle engine, to give the same number of impulses, would require 32 crankpins.

This example is merely one way of illustrating the great compactness and simplicity of the engine which is the subject of this description.

This is essentially a high-speed engine, although built as a heavy duty engine. With the modern electric or gear transmission it is a desirable engine even when slow speed is desired.

Up to this time the steam engine was almost without a competitor for rail transportation. The electric locomotive is practicable only in very rare cases. In this connection it must be remembered that the automobile became a practical possibility as a result of the perfection of the internal combustion engine. There are very few steam cars in this country, while there are about 10,000,000 automobiles with internal combustion engines. There is no reason why the internal combustion engine should not be called upon to do the work for rail transportation, as it is doing now for automobile transportation. The only question is how to concentrate large powers in small space. The characteristics necessary for starting, as well as for changing speed and torque, under different circumstances can be solved in the internal combustion locomotive just as well as in the automobile. The advantages for an internal combustion locomotive are too numerous to mention. It is sufficient to mention the fuel economy, the possibility of covering tens of thousands of miles without cleaning boilers, etc. Favorable argument is too strong to let this proposition stay dormant much longer. The engine described above should be

considered, as the solution of the problem of an internal combustion engine for locomotive purposes.

While designing this engine, the conditions prevailing in this country were kept in mind all the time.

The automobile became the most popular machine in America because it does not require any expert or licensed engineer to operate it. The whole mechanism is locked up in boxes and the driver is given very few controls which he can master in an hour or so.

The above described engine was built with the above mentioned purpose in view, viz., to build an engine with the minimum amount of mechanism and designed so that it could be built on the interchangeable principle and be turned out in large quantities at a low figure.

Further, it is designed so that there are no adjustments required, and two control handles are all that is necessary for starting, operating and running. The whole engine, with the exception of beds and crankshafts, is built up of small parts so that in case of defective parts same could be replaced instead of repaired.

This engine has been built in a 1,000 H.P. unit, which is made in four sections. A six section unit would give 1,500 H.P. A. 10,000 H.P. engine could be made on this system with parts, the dimensions of which are not new to the art and therefore could be built on sure lines.

BOLINDER'S CRUDE OIL ENGINES

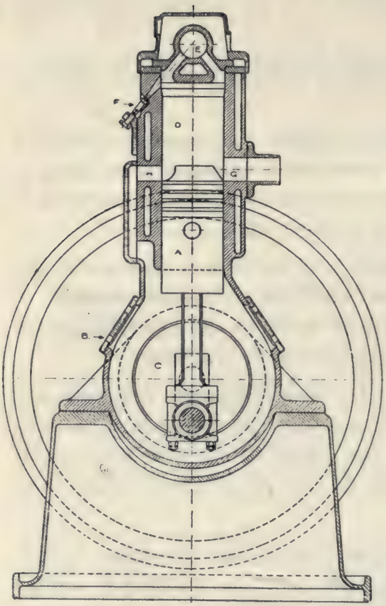
(RUNDLOF'S PATENTS)

STATIONARY AND MARINE ENGINES

The Bolinder engine is essentially a typical light weight engine following the general principle identical to all two-cycle machinery of the Semi-Diesel type. In general construction the engine is exceedingly simple in design and accessible throughout.

The manufacturer of this engine, the J. & C. G. Bolinder Co., Ltd., at Stockholm, Sweden, have added some improved features, particularly in the marine type, which makes the engine very desirable on ships operating by auxiliary power, where weight factors are of vital importance and elimination of space of necessity calls for Internal Combustion power-producing machinery.

The consumption of fuel compares well with others of this type of engine. For fuel most any residue oil can be used. The specific gravity of fuels should be preferably about 0.88 and the heat value about 10,000 calories per kilo or 18,000 B.T.U.'s per pound.



Demonstration of "Cycle of Operation," Bolinder Two-Cycle Semi-Diesel Stationary Engine.

In following description of the working performance of stationary engines of the Bolinders, an accurate idea will be formed of the principle underlying the two-cycle type.

When the piston (A) at the end of its outward stroke is moving in towards the ignition chamber (E), the necessary air for combustion is drawn in through the air valves (B) into the enclosed crank housing and at the same time, the air in the cylinder (D) is compressed.

When the piston (A) has reached its extreme inward position, a certain amount of crude oil is injected into the ignition chamber (E) through the nozzle (F), and the fuel charge explodes the expanding gas, driving the piston outward towards the shaft.

During this outward stroke of the piston, the air in the crank housing is compressed. As the piston nears the end of its stroke, the exhaust port (G) opens, and immediately after also the inlet air port (H).

The burnt gases escape by the exhaust port (G), while the compressed air in the crank housing entering the cylinder by the port (H), completes the scavenging work, and furnishes the cylinder with the air necessary to make up the next fuel charge.

It will be noticed that the ignition chamber (E), has two ports; by this means it is blown through with fresh air every revolution, an important feature for securing a rapid and effective ignition.

The piston is now on the inward stroke again and the cycle is completed.

Starting Engines with Compressed Air: Larger engines and all engines having more than one cylinder have a special starting arrangement consisting of an air receiver fitted with pressure gauge and stop valve connected by a pipe to a valve on the cylinder. Starting the engine by means of air pressure is accomplished as follows:

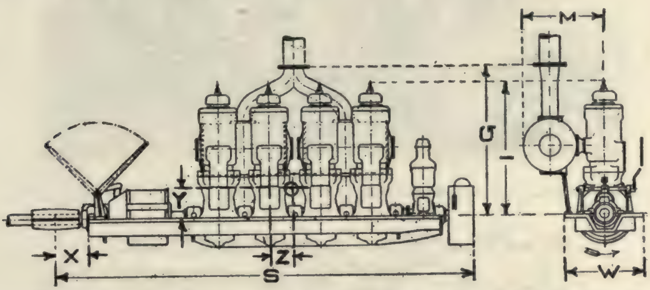
After the engine bulbs have been sufficiently heated by means of the blow lamps, the blow-off cocks on the cylinders are opened and the flywheel is turned over until the piston in the cylinder to which the starting device is attached has just commenced its downward stroke, after which the blow-off cocks are closed again.

The blow-off valve on the starting device is now closed, the stop valve opened, and the hand wheel opened up two or three complete turns.

After fuel has been injected into the cylinders by a couple of good strokes of the fuel hand levers, the starting valve is opened quickly by means of the hand lever and is held open about half a second, allowing the pressure in the air receiver to set the engine running. As soon as the engine starts, the starting valve is closed quickly, the hand wheel screwed down, and the stop valve closed while the blow-off valve is opened to allow the remaining gas in the pipe and valve to blow out, as otherwise the starting valve may show a tendency to stick.

As soon as the engine is running normally, the air receiver is loaded as follows:

After the blow-off valve has been closed, the stop valve to the air receiver is opened, after which the loading valve is opened and the pressure allowed to build up in the receiver until the pressure gauge shows from 8 Kg. to 12 Kg. pressure above the atmosphere—equal to 120 lbs. to 180 lbs. per square inch.



Plan View of Reversible Type of Bolinder Marine Engine.

The loading valve is now closed as well as the stop valve. Lastly, the blow-off cock is opened to allow the gas remaining in the pipe to come out. The air receiver should always be kept under pressure. The spring on the starting device should be adjusted so that the valve does not open by the pressure in the cylinder.

In the maneuvering of the direct reversible engine, following directions should be strictly adhered to:

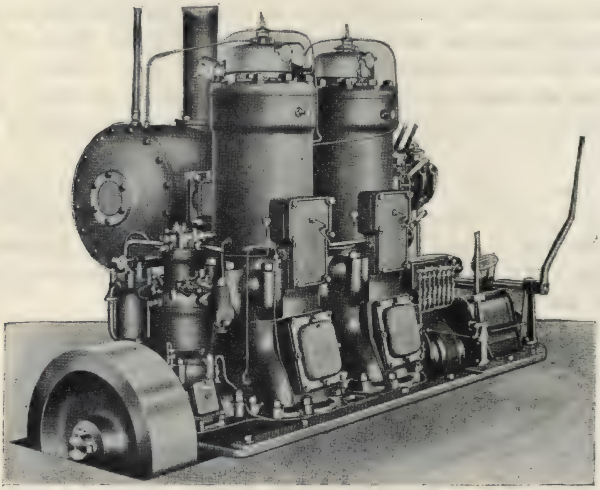
It is understood, that the reversal direction of rotation is effected by means of pre-ignition without appealing to any external source of power such as compressed air, electric, etc.

1. The clutch is thrown out by means of the hand lever.

2. The reversing lever is pulled out aft (for going astern). This movement causes the engine instantaneously to slow down; a charge of oil is automatically injected at the appropriate stage of the cycle, and the movement of the piston is immediately reversed.

3. The reversing lever is returned to its central position.

4. The clutch is thrown in again. The whole maneuver is performed by two hand levers. To change from astern to ahead, the procedure is exactly the same, except that the lever is thrown over in the opposite direction. The reversing should not be moved until the clutch has been thrown out; but should then be held either at astern or ahead, until the engine has reversed after which it can be returned to its central position.



150 H.P. Bolinder Crude Oil Engine, Two-Cycle.

STARTING WITH COMPRESSED AIR.

In the following illustration, a good view is allowed demonstrating the usage of compressed air, as used on the Bolinders type, for starting purposes. On large engines and all engines having more than one cylinder are fitted with a special starting arrangement consisting of a air receiver (102) fitted with a pressure gauge and stop valve (103) connected by a pipe to a valve (104 A) on the cylinder.

To start the engine by compressed air is accomplished as follows:

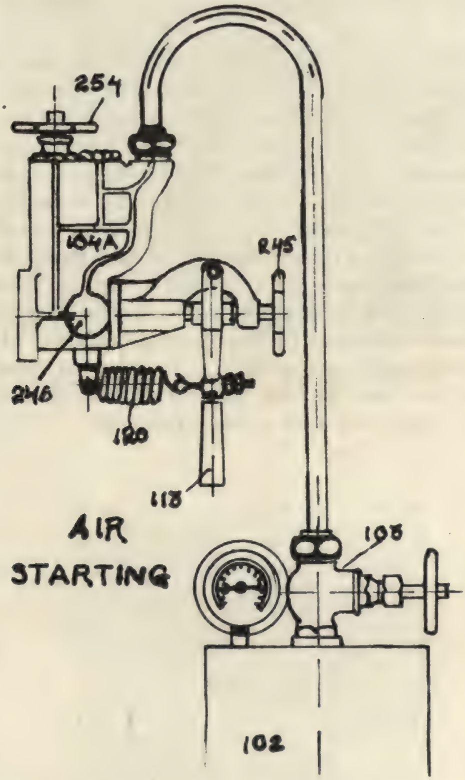
After the ignition bulbs have been sufficiently heated by means of the blow lamps the blow-off cocks on the cylinders are opened and the flywheel is turned over until the piston in the cylinder to which the starting device is attached has just commenced its downward stroke, after which the blow-off cocks are closed again.

The blow-off valve (246) on the starting device is now closed, the stop valve (103) opened, and the handwheel (245) opened up two or three complete turns.

After fuel oil has been injected into the cylinders by a couple of good strokes of the fuel hand levers, the starting valve is opened quickly by means of the hand lever (113) and is held open about half a second, al-

lowing the pressure in the air receiver to set the engine running. As soon as the engine starts, the starting valve is closed quickly, the hand wheel (245) screwed down, and stop valve (103) is closed while the blow-off valve (246) is opened to allow the remaining gas in the pipe and valve to blow out, as otherwise the starting valve may show a tendency to stick.

As soon as the engine is running normally the air receiver is loaded as follows:



Demonstration of Air Starting Method on Bolinder Engines.

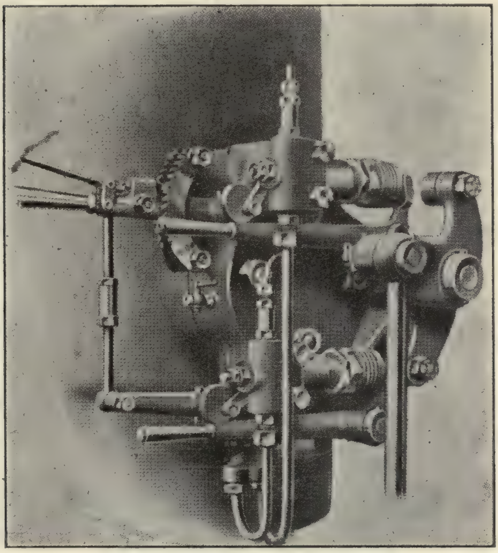
After the blow-off valve (246) has been closed, the stop valve (103) to the air receiver is opened, after which the loading valve (254) is opened and the pressure allowed to build up in the receiver until the pressure gauge shows from 120 lbs. to 180 lbs. per square inch, equal to from 8 Kgs. to 12 Kgs. above atmospheric pressure.

The loading valve (254) is now closed as well as the stop valve (103). Lastly, the blow-off cock (246) is opened to allow the gas remaining in the pipe to come out.

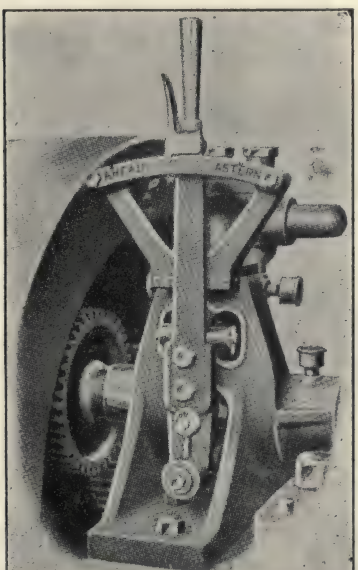
The air receiver should always be kept under pressure. The spring (120) on the starting device should be adjusted so that the valve does not open by the pressure in the cylinder.

The accompanying illustrations pertaining to the Petter Crude Oil Engine, built in England, demonstrates the simple method of reversing arrangement. This engine, which is of the hot surface two-cycle type, is built in sizes having four cylinders up to about 300 H.P. The operation of the pump as shown in the illustration, follows the system of direct reversal of rotation of the crankshaft, and is explained in the following paragraph.

By manipulation of the hand lever, shown in illustration, the fuel pump is placed out of action when so desired. With the slowing down in speed of the engine the reversing lever is moved to the "astern" position, which movement allows compressed air to be admitted to the cylinder on the upstroke of the piston. The volumetric efficiency of the engine, owing to its excellent construction, equals engines of most up-to-date types. It is principally due to experiences gained by engineers of this company that the accomplishment of this pump ranks as an elegant mechanism on this machine. The reversing of this engine may be accomplished without stopping the engine. The pressure in the cylinder causes the piston to descend in the reverse position without stopping the engine, as previously mentioned. After this has been accomplished the fuel pump is then allowed to be in operation again and the reversing lever is returned to the center or neutral position.



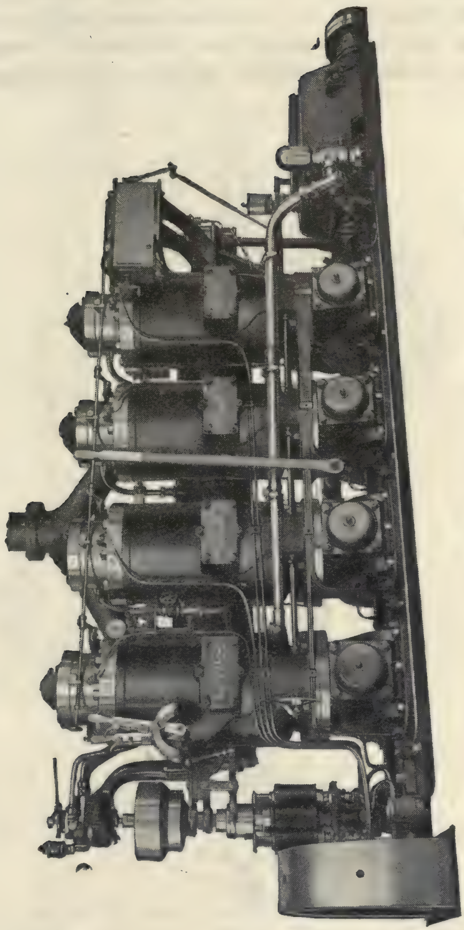
Fuel Pump Arrangement on Petter Crude Oil Engine.



Maneuvering Lever on Petter Crude Oil Engine.

STANDARD SPECIFICATIONS OF BOLINDER'S HEAVY OIL ENGINES.

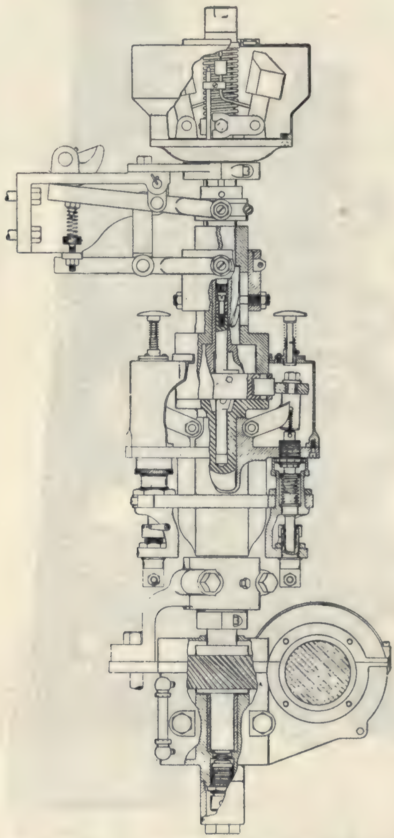
B. H. P.	90	120	160	240	320	500
Revolutions per minute	300	250	225	250	225	160
Weight, net in cwts.	112	173	230	337	456	1016
Fuel consumption, lbs., per B.H.P. per hour	0.61	0.58	0.58	0.58	0.58	0.55
Vertical C. G. Dimensions—Y, inches	13	15⅝	17¾	14½	17¾	22½
Horizontal C. G. Dimensions—Z, inches	8	9	10⅝	8¼	13	13½
Dimensions of engines, in inches:						
Length—S	130	147	175	236	266	322
Width—W	38	43	49	47	51	71
Width—M	34	38	52	38	52	72
Height—I	65	76	86	76	86	121
Height—G	--	--	--	88	97	132
Length—X	9½	5¼	13¼	8½	14	10¾
Diameter, in inches:						
Fly wheel—D	27¾	33½	36½	35½	39½	55
Propeller, 3 blades	47½	59	67	67	79	102
Propeller shaft	4.921	5.708	6.3	6.69	7.48	10.04
Volume, cubic feet	424	600	813	1130	1480	3310
Weight, cwts.	140	206	287	417	540	1179
Weight, kgs.	7500	10800	15600	21200	28000	62000
Type	R-300	R-400	R-500	R-400	R-500	R-600



Kahlenberg Heavy Duty Crude Oil Engine

THE KAHLENBERG ENGINE

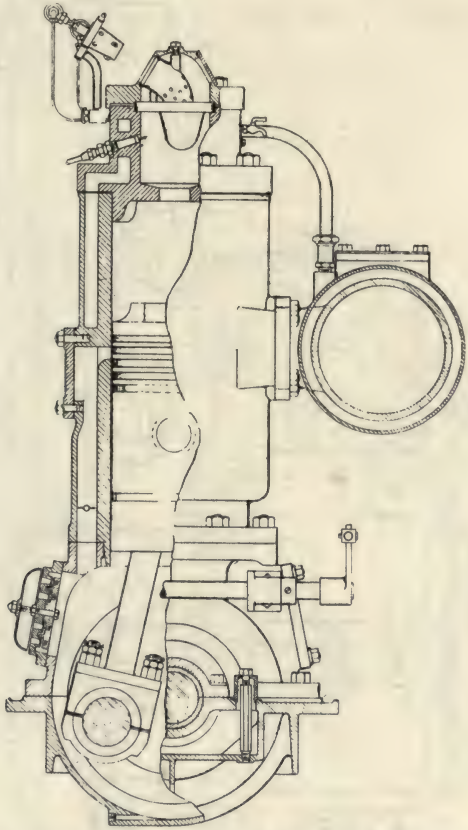
To those more familiar with the more modern types of semi-Diesel engines, it will be seen at a glance that the Kahlenberg is entirely different. There are numerous features in the design and construction to draw the attention. Notably the levers on the forward end of the machine—one is the speed and the other injection control. Speed control is the same as the throttle valve of a steam engine and with it any speed from just turning over to the rated speed of engine is obtainable, without relighting the torches and without missing a single impulse from no load to full load.



Valve Arrangement, Gears and Governor Equipment on Kahlenberg Oil Engine.

Inasmuch as the fuel delivery is instantaneous and is at the proper time relative to the piston traveling over the top center, no water injection is used. With the injection control lever, the fuel injection is made to occur at the proper time for best operating results, and is adjusted while engine is running.

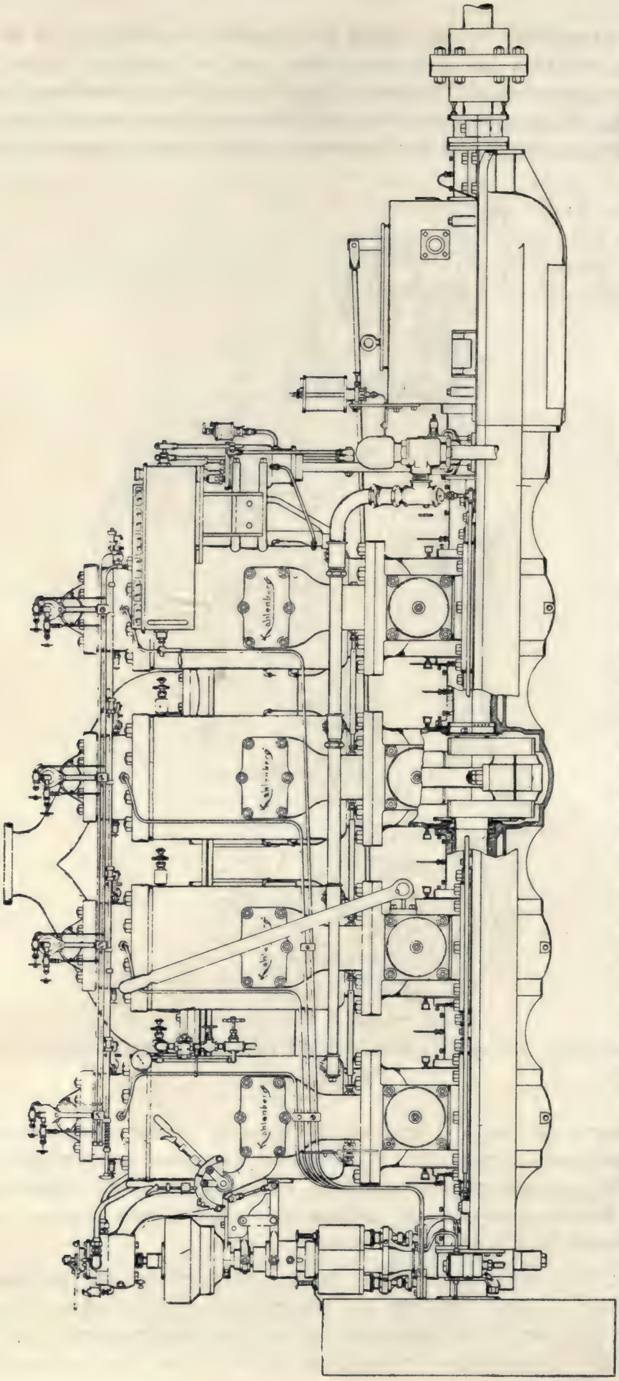
The reversability of the engine is a feature which deserves mention. Kahlenberg engines are reversible and can be operated either direction. The engine can be reversed from forward to the backward motion and the time of fuel injection can be advanced to any part of the stroke, both on the go-ahead and the reversing, while the engine is in operation.



Partial Section Through Cylinder and Bearing of Kahlenberg Marine Oil Engine.

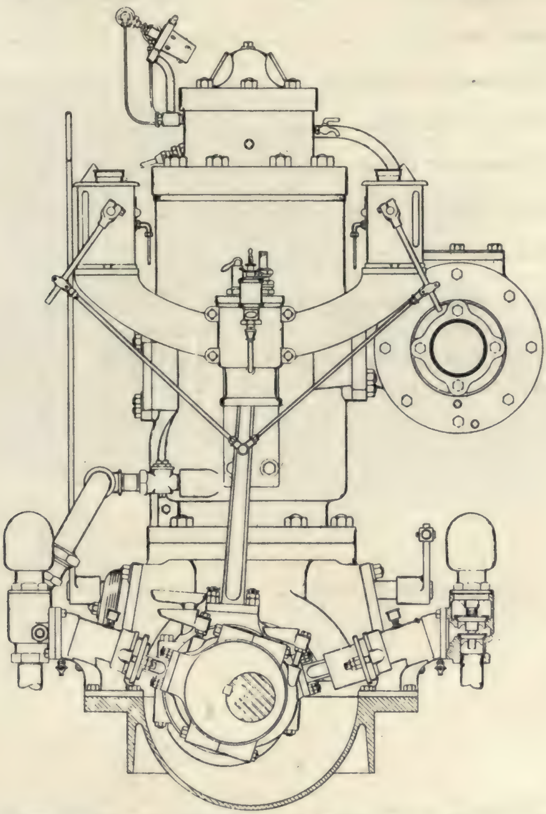
On nearly all surface ignition engines, as noted in the description of the numerous engines, the time of fuel injection is permanently set, requiring a stoppage of the engine to make adjustment. This is not to imply as disadvantageous, but rather as a fact corresponding with the general design based upon the respective type.

In the illustration pertaining to the governor of the vertical type, the similarity to those used on full Diesel engines is notable. The sensitiveness of this specific arrangement allows the engine speed to he held at the number of revolutions at which it is set, with just the least perceptible



Diagrammatical View of Kahlenberg Marine Oil Engine. This engine is direct reversible. The engine is well adapted for service where a machine of rigid construction is called for to perform heavy duty work.

variation from no load to full load. It acts on the driving cam, which in turn acts on all the fuel pumps, thus changing their stroke instantaneously, regardless of position when the change in load occurs. The maximum R.P.M. that the governor will hold the engine at can be varied while the engine is in operation by a knurled screw spring adjustment. Thus, if the main governor spring is adjusted for 350 R.P.M., the small knurled screw under the fork lever of governor may be adjusted until desired speed is obtained.



End View Through Eccentric Pit of Kahlenberg Marine Oil Engine. Note the Double-Set of Pumps.

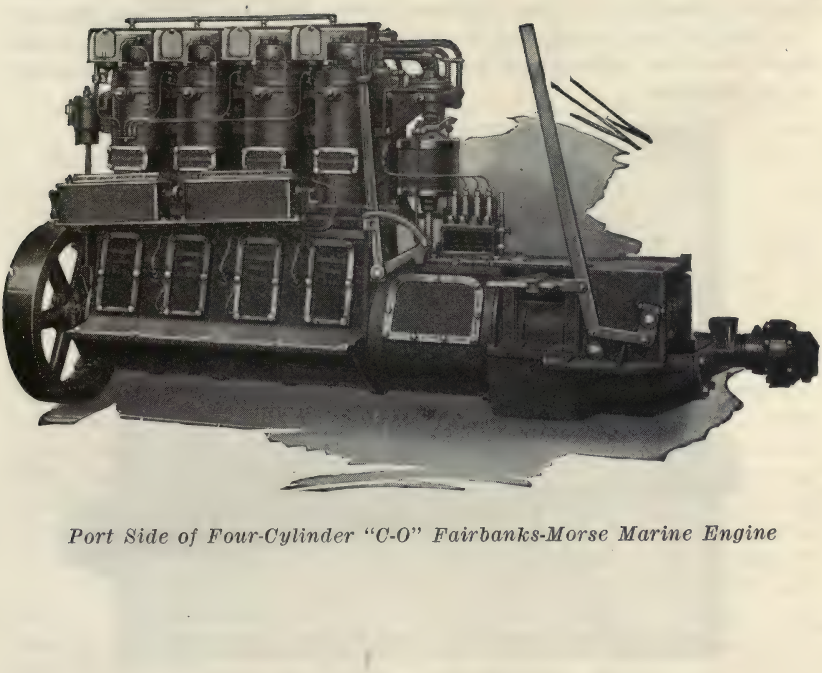
The air compressor attached to the engine is of the single stage type and furnishes air for starting at 150 lbs. pressure. It unloads itself automatically when up to this pressure and has also a hand unloading device so the compressor can be stopped or started as required and the Hand Air Control is located in intake ports and all are connected to one lever (below throttle and injection controls). It is used only when operating slow for long periods and is then closed to prevent in-rush of cold air cooling the hot bulbs.

SPECIFICATIONS AND DIMENSIONS OF KAHLENBERG MARINE OIL ENGINES

HorsePower Capacity 100 H.P. to 120 H.P.

Size of Cylinders	10×10½ in.
Diameter of Balance Wheel	36 in.
Diameter of Crank Shaft	4 in.
Length of Main Bearings	10 in.
Length of Intermediate Bearings	8¼ in.
Extreme Length of Engine	12 ft. 9½ in.
Height from Center of Crank Shaft to Top of Engine	4 ft. 3 in.
Distance from Center of Engine to Outside of Silencer	23½ in.
Width of Bedplate	26½ in.

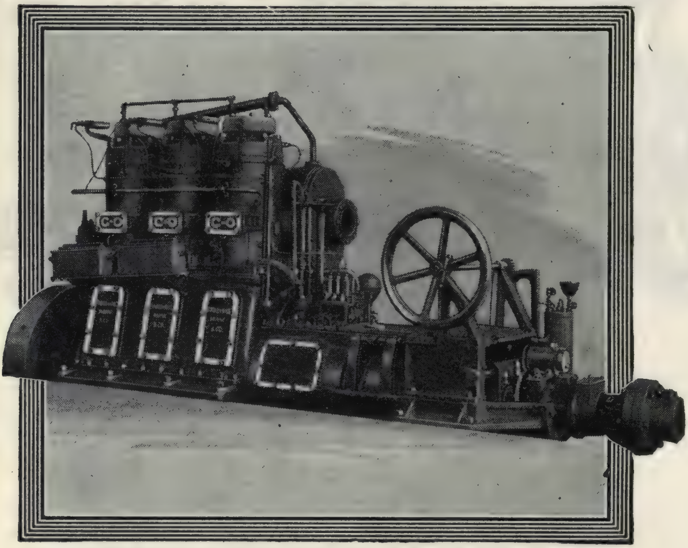
Note.—Engines of above specifications and capacities are of 4 cylinder construction, reversible in addition to reverse gear. The engines will operate on fuel oil of not less than 24 degrees Baume. Consumption five-tenths pounds per B.H.P.



Port Side of Four-Cylinder "C-O" Fairbanks-Morse Marine Engine

FAIRBANKS-MORSE "C-O" MARINE OIL ENGINES

Principle of Operation: Engines of this type operate on the two-stroke cycle. Following is the procedure of power production: On the upward stroke of the piston air is drawn into the crankcase through a set of automatic valves located in the crankcase and on the downward stroke this air is slightly compressed. Near the end of this stroke the exhaust ports are uncovered by the piston, permitting the burned gases in the cylinder to escape. Shortly after the exhaust ports are opened, the piston uncovers the air ports on the opposite side of the cylinder and the air compressed in the crankcase rushes into the cylinder, cleaning the latter of burnt gases and charging it with fresh air. After the piston closes the inlet and exhaust ports this charge of pure air is compressed in the cylinder.



Fairbanks-Morse "C-O" Engine, Marine Type

Just before the piston reaches its upper dead center, the fuel is injected in the form of a fine spray. At the dead center, when the compression has reached its maximum, ignition automatically takes place and the resulting pressure drives the piston downward, doing useful work. After expansion is completed, the piston again uncovers the exhaust ports and the cycle of operation is repeated.

Air Seal in Crankcase: Fresh air for combustion enters the cylinder at the end of the down stroke under slight pressure from the crankcase. It is important that compression be maintained and equally as important that the lubrication oil shall not be blown out of the bearings. This is accomplished by a unique arrangement without stuffing boxes and without putting any pressure on the main bearings.

Combustion Chambers: The "C-O" is not a hot bulb engine. The combustion chambers are almost entirely water cooled, with the walls of sufficient thickness and strength to meet all demands of the service.

Exhaust Manifold: The exhaust manifold is of large capacity and acts as a muffler. It is closed at the ends with removable heads, each of which is provided with a pipe connection for the exhaust outlet. A cleanout plate is provided opposite each cylinder, which also serves for inspection of the exhaust ports. The manifold is completely water jacketed and all the cooling water after leaving the cylinders, flows through the jacket of the manifold.

Circulating Water Pump: This pump is of the plunger type and is of large capacity. It is located at the after end of the engine, driven directly from the crankshaft by means of an accentric.

Governor: The governor for all engines up to and including 100 H.P. is mounted in the flywheel, and for all larger engines directly on the crankshaft. It is of the centrifugal type and acts on the fuel pumps, automatically altering the stroke of the pump plungers according to the power required from the engine.

Speed Control: Each engine is provided with a speed control which makes it possible to run the engine at any pre-determined rate, from full down to 35 per cent of its rated speed.

Fuel Injection Pumps: There is no part that requires more careful and accurate machining than these pumps. The plungers are of steel, carbonized, hardened and ground. The suction and discharge valves are contained in cages so that they can be removed and replaced in complete, self-contained units. There is one pump for each cylinder and each pump can be held out of action or even replaced while the engine is running without interfering with the operation of the remaining cylinders and pumps. The pumps are operated by cams, which are of steel, carbonized, hardened and accurately ground by a special cam grinder and are of such shape that all levers and rods operating the pumps are always in contact with each other, thus insuring quiet operation. The cams are provided with adjustments so that the timing of the fuel injection may be advanced or retarded if necessary.

Air Compressor: An air compressor, used to pump air for starting the engine and such work required by air, is attached to the engine. Is of the single-action, single-stage type, driven by an eccentric directly from the crankshaft. The entire compressor and head are water cooled. The suction valves are of the poppet type, made of steel and mounted in removable cages, being so constructed that they may be held off their seats when the tanks are filled to the desired pressure. The discharge valves are of the cup type, made of steel, mounted in removable cages. The entire cylinder head is removable.

Electric and Torch Starting: Each engine is equipped with an electric device to initially heat the combustion chambers. This is necessary when the engine is started from a cold condition. The electric outfit consists principally of a storage battery, charging generator and one

ignition or heating plug for each cylinder. The plug carries an element that can quickly be made red hot by current from the battery. The element can be replaced at small expense with renewing the entire plug. With this device an engine can be started in 30 seconds. As an auxiliary means, kerosene torches are also furnished for heating the combustion chambers.

Reverse Gear: All engines up to and including 100 H.P. are equipped with reverse gears. The forward drive consists of a multiple disc clutch, the discs being faced with a special non-burning, asbestos woven material. The reverse drive consists of a set of gears and pinions, enclosed in a drum and running in oil.

Air Starter: All engines are started with compressed air (Engines above 100 H.P. are direct reversible and will be described later). A distributor mounted on the after end of the engine delivers air to each cylinder in proper rotation. The distributor comprises a set of valves, one for each cylinder, operated by a cam located directly on the crankshaft. The distributor is connected to a tank holding compressed air. It is in operation only during starting, and all parts stand still when the engine is in regular operation.

Direct Reversing Engines: All engines of 150 H.P. and larger are started and reversed with compressed air. The necessary air for each engine is stored in tanks which are pumped up by an air compressor attached to the main engine and driven by an eccentric. A separate air compressor driven by a small engine is in some cases found desirable, as the air can be stored then independently of the larger engine. The starting and reversing mechanism consists essentially of a housing containing two rotating discs of which one is timed for ahead and the other for astern. Each disc has a slot which admits air to the pipes leading to the cylinders in order for rotation in the proper direction. A hand controlled air inlet valve admits air to the ahead or astern disc at the will of the operator. The distributor is positively driven from the crankshaft by a set of gears.

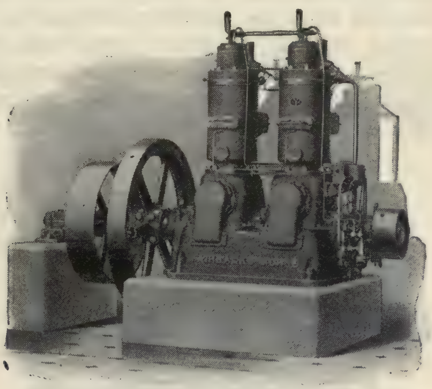
One important feature in reversing liquid fuel engines is the control of the fuel; that is, the injections must cease positively as soon as the control lever is moved, and the fuel also must be turned on immediately the engine runs in the desired direction. This is accomplished by cams and levers controlling the suction valves of the fuel pumps, allowing the fuel to be by-passed back into the oil tank or pumped into the cylinders as required. All movements controlling the air as well as the fuel are accomplished with one lever.

In accompanying illustration of the 75 H.P. "C-O" engine, it will be noticed that the accessibility is a distinctive feature highly commendable. The crankcase is of the completely enclosed type. It has heavy supporting flanges running along its entire length. It is partitioned off into as many compartments as the engine has cylinders, and each compartment is provided with a drain for the surplus lubricating oil. A large opening, covered with a plate, is provided on each side of each com-

partment for the purpose of removing the connecting rod box and intermediate bearings. On the inner side of the plates are the special ring-shaped scavenging air intake valves, giving very large openings with very small lifts, therefore having very high efficiency.

The main bearing shells are of cast iron, provided with dovetail grooves to hold a high grade bearing metal. They are rigidly supported in the lower part of the crankcase casting, and can be taken out without removing the shaft.

The engine is adaptable for auxiliary purposes abroad ships depending upon sail power, tug boat service, ferry boats, etc., its special features may be summarized in following: 1. The ability to use a wide variety of low priced fuel oils. 2. Operation without water in the cylinder. It is possible to use water with the fuel to slightly increase the power of an engine, but it results in certain and rapid wear on both cylinder and piston. In other words, a higher rate is assured by mini-



75 *H. P. "Y" Oil Engine*

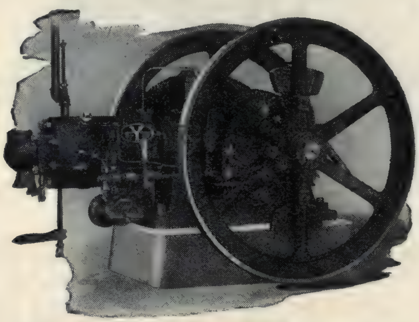
mizing the use of water. 3. Perfect lubrication. Fuel is injected into the combustion chamber—not into the cylinder where it would impair lubrication. The fuel never comes into contact with the lubricating oil. The main bearings, pistons, piston pins, and crank pins, are lubricated from a force feed pump. 4. No excessive temperature. There is no hot ball or hot bulb to overheat or burn out. The combustion chamber is water jacketed and its temperature is thereby always under control. 5. Air tight crankcase. Special air seals on the crank shaft are used instead of stuffing boxes that would require repacking. With this construction the end main bearings are not enclosed in the crankcase; thus lubricating is not interfered with by air pulsations and the bearings may be inspected readily. 6. A special quick starting arrangement.

As only a comparatively small amount of heat in the combustion chamber is necessary for starting, and since the "C-O" is provided with special arrangements for quick starting, it does not require a thin wall hot bulb. On the contrary, the walls of the combustion chamber are quite heavy; hence the necessity of replacement is very remote.

FAIRBANKS-MORSE "Y" VERTICAL OIL ENGINES

The "Y" engines are adapted to any stationary power purpose within their range of sizes. They can be belted direct to line shafts for driving factories, cotton gins, elevators, flour mills, etc., or they can be belted direct to an individual machine, for example an air compressor, ice machine, pump of any variety, or any similar unit.

For the generation of electric current, which may be used for lighting or for power, "Y" engines can be furnished either for belting to the generators or for direct connection to them. The choice between these two varieties will depend upon the condition of the plant as to space and other items. Whether for belt or direct connection, the engines are equipped with flywheels of sufficient weight to prevent pulsations or flicker in electric lights from the generators, and they may be arranged for parallel operation or two or more similar units on the switchboard.



Typical Horizontal "Y" Oil Engine

Principle of Operation: The "Y" engine operates on the two-stroke, moderate pressure principle, with the fuel injection into the combustion space by means of a simple pump, properly timed, and controlled by the governor in proportion to the load on the engine.

The "Y" is not a hot-bulb engine, as the combustion chamber is entirely water jacketed. There are no hot plates or firing pins, but the heat remaining in the combustion chamber, together with that produced by the compression of the charge of air, ignites the oil, which burns with more of an expansive pressure than the explosion in the ordinary internal combustion engine.

This system of combustion is a development accomplished by experiments of many engineers and represents an advancement in this respective type of engines.

A Comparison of Types: Within the range of sizes in which they are built, the "Y" engines have several marked advantages worthy of mention.

The "Y" engine has an initial compression of scarcely more than half that of the Diesel type and does not have the extremely high firing pressure of the latter. This allows the use of fewer piston rings (constituting the largest element entering into engine friction), and the "Y" engine operates with proportionately less pressure on all the main working parts, such as pistons, rods, crankshaft and bearings.

By using the solid injection principle, this type does not require a two or three stage high pressure air compressor, which in this case would be an expensive auxiliary to maintain and in which the Diesel absorbs from 10 to 15 per cent of the total power generated.

There is an absence in this engine of all inlet and exhaust valves, which in the four-cycle type require great care in maintenance and renewals throughout the life of the engine.

Owing to the lower working pressures and temperatures of the "Y," the upkeep is extremely low in comparison to engines where mechanical arrangements are of a nature which require considerable upkeep expenses.

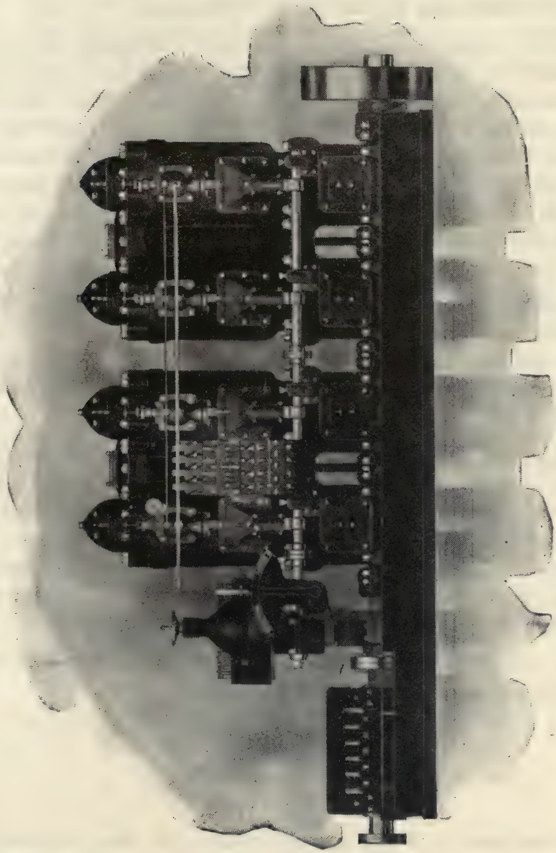
DIMENSIONS OF 30, 45 AND 60 H.P. "C-O" ENGINES

(Fairbanks-Morse Marine Type)

Horsepower	30	45	60
Number of Cylinders	2	3	4
Revolutions per minute	400	400	400
Net weight of engine with rev. gear...lbs.	5050	6870	8000
Length over all	8′ 10¼″	9′ 11½″	11′ 0¾″
Width over all	3′ 6½″	3′ 6½″	3′ 6½″
Height above base flange	4′ 6½″	4′ 6½″	4′ 6½″
Depth below base (without flywheel)	9½″	9½″	9½″
Flywheel — Diameter	33″	33″	33″
Flywheel — Face	6″	6″	6″
Flywheel — Weight	600 lbs.	600 lbs.	600 lbs.
Water Inlet Pipe	1″	1″	1″
Water Outlet Pipe	¾″	¾″	¾″
Fuel Inlet Pipe	½″	½″	½″
Exhaust Pipe	5″	5″	5″
Diameter of Propeller Shaft	2″	2½″	2½″
Diameter of Propeller (approx.)	34″	38″	42″

Note.—On larger types of engines of "C-O" Fairbanks-Morse, following figures are given in regards to dimensions of propeller shaft and diameter of propellers:

	75 and 100 H.P. (approximately)		
Diameter of Propeller	44″	50″	
Diameter of Propeller Shaft	3½″	3½″	
	150, 200 and 300 H.P. (approximately)		
Diameter of Propeller	66″	72″	78″
Diameter of Propeller Shaft	6½″	6½″	6½″



240 H. P. Direct Reversing Gulowsen-Grei Marine Engine

GULOWSEN GREI MARINE HEAVY DUTY ENGINE

The Gulowsen Grei Engine is of two-cycle construction. It is of Norwegian design and built since 1919 in Seattle, Washington. There are features of exclusive distinction on this engine, which have proven highly satisfactory in operation of this type of machine.

The engine, which is equipped with an electric heating device, which takes place of the torch usually employed in accomplishing its initial starting temperature, can be started in 20 seconds.

This device consists of a wire element contained in a plug, and exposed to the oil spray in the combustion chamber. The element is heated by current from a 6-volt storage battery. When the engine is revolved by means of compressed air, the oil spray from the injection nozzle comes in contact with the heated wire elements, ignition takes place and the engine starts in motion. After the engine has been running for about one minute, the combustion chamber becomes heated sufficiently to ignite the oil without the heat of the wire elements, when the circuit between the plug and the battery may be broken.

These engines operate on crude or fuel oils of either asphalt or paraffine base, containing 18000 B. T. U.'s per pound or more, and having a gravity of not less than 24 degrees Baume. They also operate on all higher gravity oils such as gas oil, solar oil, stove distillate, etc. A sulphur content of not over 1 per cent is permissible.

Water injection cannot be used with asphalt base, such as Mexican or California oils, on account of the chemical action which takes place in the cylinder. However, with paraffine base oils, an increase of power of about 15 per cent may be obtained when necessary by using water injection. These engines run as well without water injection, and where it is not convenient to carry water, it is not used. It is practice in Europe to carry a tank of water for reserve power.

The fuel consumption is about .5 lb., or .066 gal. per B.H.P. hour, depending entirely on the heat units in the fuel.

Construction Details

Base: The base is of the channel type, very deep and heavily constructed, which gives a maximum strength and rigidity. This is very necessary, especially where the engine is installed in a wooden vessel which will weave in a seaway, as under severe conditions a weakly constructed, shallow base, will crack in the center.

The main bearing housing are turned to fit the main bearings, which are babbitt lined cast iron shells. When the shells are of a uniform thickness, the alignment of the crankshaft will be correct. Each crank pit is separate from the rest and has drain pipes with check valves to drain any surplus lubricating oil.

Crankshafts: The cranckshafts are cut from solid billets of high carbon open hearth steel of high tensile strength, and conform to Lloyd's specifications. All crankshafts are exceptionally heavy, have a bearing between each throw, and have very large bearing surfaces. Each crankshaft throw on the engines up to 240 B.H.P. has a pair of counterweights, doubly fastened by means of keys and bolts.

Thrust Shaft and Bearings: Thrust shafts are also cut from steel forgings of high tensile strength. The thrust bearings are horse shoe jokes, the type used in marine steam engine practice. The jokes are unusually large and are removable for inspection or re-babbitting without disturbing the thrust shaft. On all four-cylinder engines, 125 H.P. and up, the jokes are water cooled.

Crankcases: Crankcases are cast separate from the cylinder and are of ample strength to insure rigidity under all loads. The hand hole plates on each side of the crank chamber are made very large for easy inspection and removal of the connecting rod bearings. On the inner side of the plates are the scavenging or intake valves, made of leather with an alloy steel spring.

Combustion Chamber Head: The combustion chamber head is half water cooled and half air cooled. The air cooled part never gets hotter than a black heat.

Fuel Injection Pumps: The fuel injection pumps are constructed with phosphor bronze bodies and steel plungers, hardened and ground. All pumps are subjected to a hydraulic test pressure of 5000 pounds. The plungers are long and work with an oil seal. Each cylinder has an individual fuel pump.

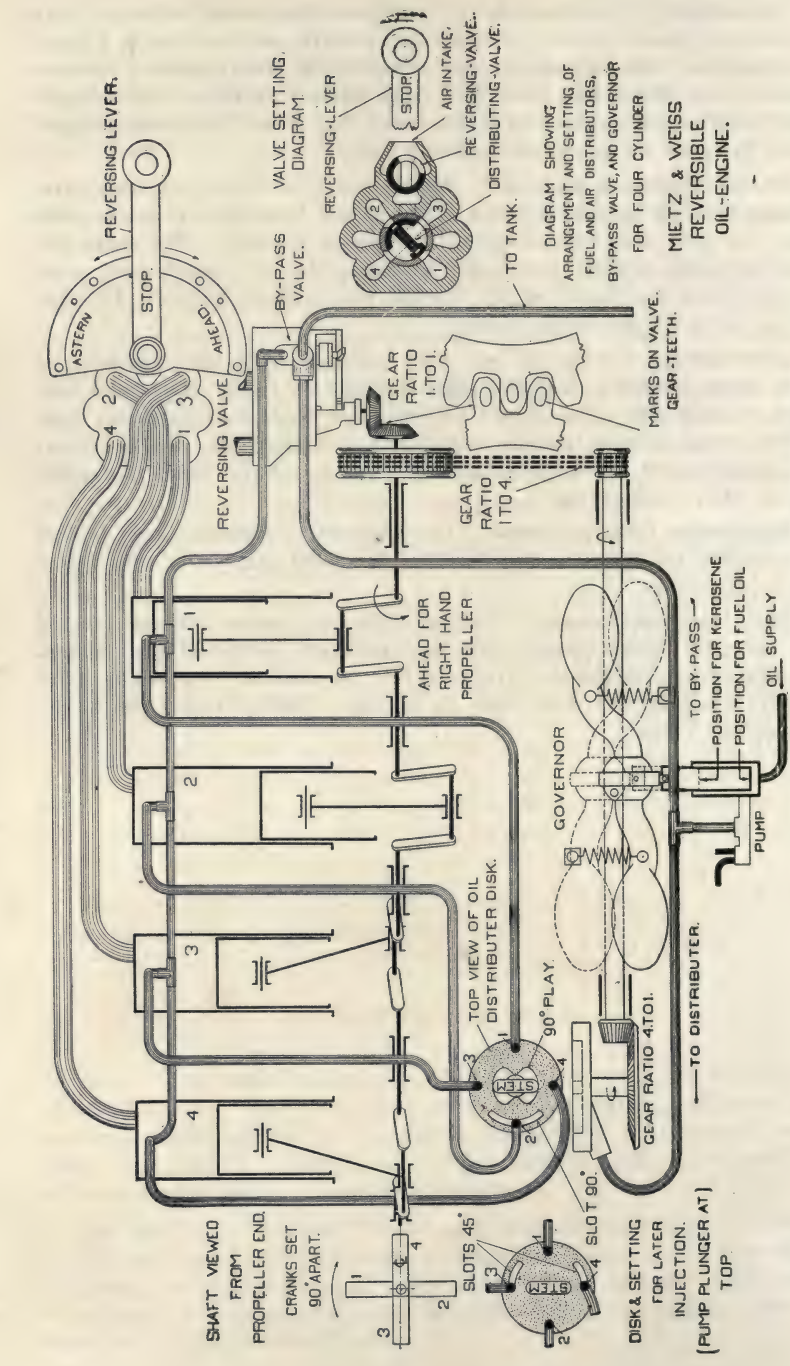
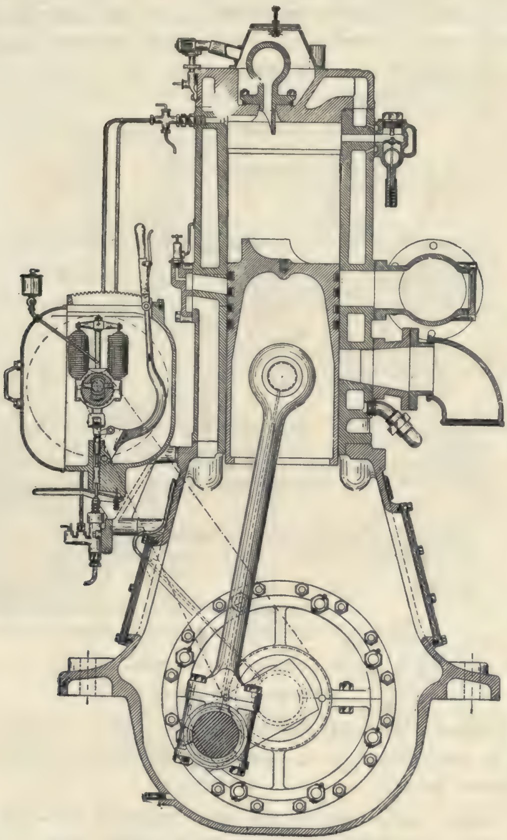


Diagram Showing Arrangement in Installation

MIETZ & WEISS OIL ENGINES.

Like most Ignition Surface engines or such where hot balls, hot tubes, etc., are used, this engine is of the two-cycle construction. Since the fuel is sprayed directly in the combustion chamber shortly before completion of the compression stroke, there can be no loss of fuel through crank case leakage or through the exhaust port.

The fuel consumption of this engine is as low as .6 of a pound per horsepower hour. The engines operate on kerosene, fuel oil distillate, crude oil, or alcohol.



End View of Mietz Marine Oil Engine

The fuel is injected in liquid form through the injection nozzle at extreme high velocity.

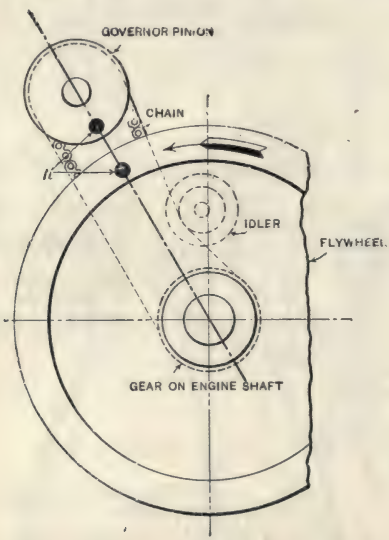
Directly in its path is the lip or tongue of the ignitor ball. The impact of the fuel against this lip is so violent that the former is atomized and scattered throughout the combustion space to form an explosive mixture. The heat of compression together with the heat of the ignitor

ball, which has been previously heated, serve to gasify the atomized fuel, and to automatically ignite the charge.

The ignitor ball is not water-jacketed. It is heated for a few minutes, before starting, by means of the burner. As soon as the engine is started, and the load is thrown on, the burners are extinguished, the ball being maintained at the proper temperature by the heat of the successive explosions.

The time of ignition is controlled and the efficiency of combustion is increased by a little water, which is injected with the incoming air through the side feed.

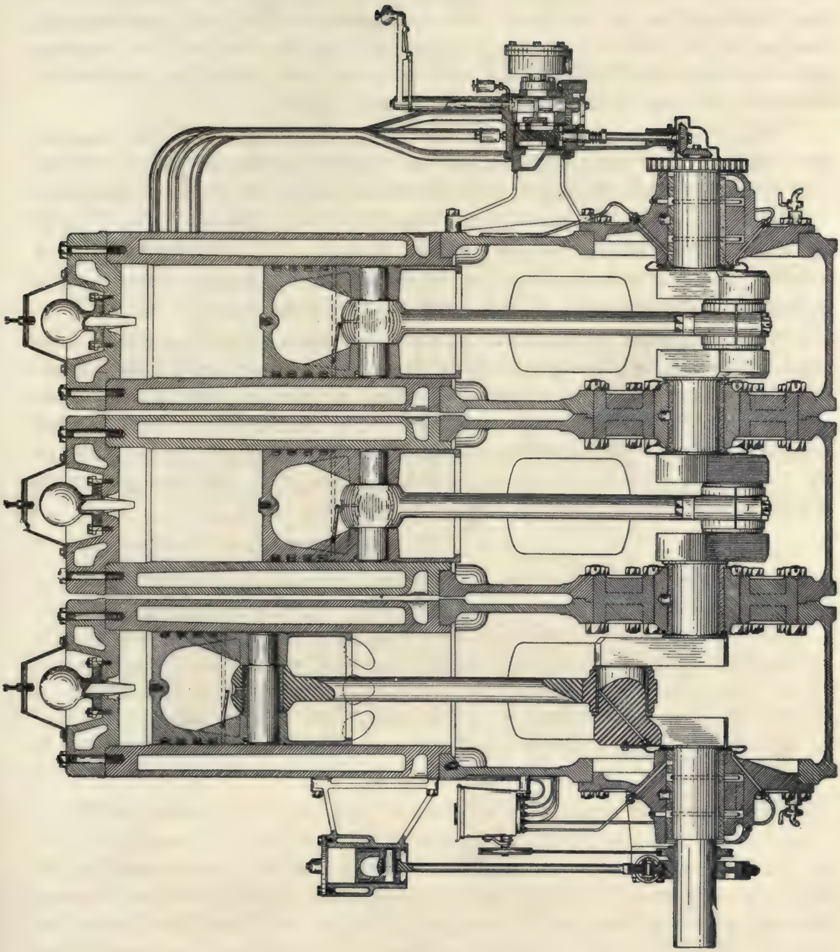
The ignited charge drives the piston down on a power stroke. Near the bottom the exhaust port is overrun. The burned gases escape and a fresh charge of air from the crank case takes their place. This is



Governor arrangement of Mietz Oil Engine, Direct Driven from Engine Shaft

compressed, mixed with fuel and ignited as before. Thus: Every upward stroke of the piston is a compression stroke. Every downward stroke is a power stroke.

A lip or tongue attached to the ball projects through the cylinder-head into the cylinder, directly in the path of the oil injection. The oil spurting from the injection nozzle striking the lip forcibly. The oil being brought in a stage of foggy substance is automatically ignited as the piston completes its compression stroke by the increasing temperature due to compression and the heat from the ignitor ball, as previously stated. For this reason the ball must be heated before the engine can be started. Kerosene torches, which will heat the ignitor ball sufficiently in five or ten minutes, are mounted on each cylinder.



Cross Sectional View Mietz & Weiss Oil Engine

After the engine is started and the ignitor ball is at a dark red heat, the torch can be extinguished. In normal operation the heat of explosion maintains the ball at an almost red heat. The temperature of the ball can be controlled by regulating the sight water feed. The damper on the air mantle should be open while the torch is burning. After extinguishing the torch, close the damper to protect the ball from the cooling effects of the air currents. The temperature of the ball also depends largely on its wall thickness and the load on the engine. The ignitor balls furnished with the engine will maintain the proper temperature at constant full load while the damper is on.

When using very heavy oils, the ignitor ball should be so inserted that the oil from the nozzle strikes the tongue on the outside (i. e., the ball turned 180 degrees about its vertical axis from the original position).

A commendable feature on the Mietz & Weiss engines is the counterbalancing of the piston and rotating parts by accurately proportioned weights which are bolted and keyed to the crank cheeks. This results in reducing the vibration of the engine to a minimum.

When stopping of engine is to be accomplished, see that there is at least 75 pounds per square inch air pressure in the starter tank. Open the water feed for a short time and push the throttle lever slowly to its off position. Close the water sight feed and shut off the fuel. In cold weather draw off the water from the jacket and pipes to avoid damage by frost. To stop a direct reversible engine it is only necessary to place the air control lever in the stop position.

CHAPTER XIII.

AIR COMPRESSORS

THE AIR COMPRESSOR

The compressing of air is not in any case as simple an operation as the pumping of water. In particular is this true where high pressures are required, involving multiple-stage compression, where factors of proportional requirements and existing conditions must be taken in consideration.

At whatever pressure the air must be delivered for the respective purpose desired, the fact remains that the mechanical efficiency on Compressor Machinery must be in proportion to work to be performed and no device known in the field of engineering requires more intimate knowledge as to its accurate performance than the Compressor.

With all our great improvements and the wonderful accomplishment in the field of engineering we may still consider the problem of machinery in its infant stage. This particularly applies itself to the use of air as a factor well worth considering.

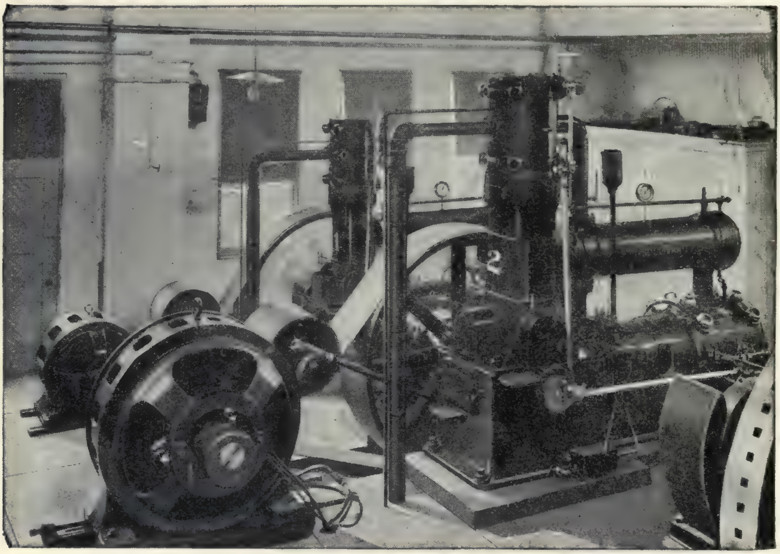
When chemically analyzing air we say that it is composed of 23 parts by weight of oxygen, and 77 parts of nitrogen. By volume the proportions are 21 parts of oxygen and 79 parts of nitrogen. We find in those figures that oxygen is somewhat heavier than air, while nitrogen is a fraction lighter, the specific gravity of the former when separated being 1.106, and in the case of the latter, 0.974, air being 1, and when liquid air is evaporated the nitrogen boils away first, which is taken advantage of for the commercial segregation of these gases.

When using the expression, atmospheric pressure, so commonly used in the engineering language, we allude to air-sphere, natural in its aspect. The air mechanically compressed brought in close confinement subject to pressure may be termed compressed air. Again after the function has been performed required of it, it again becomes "free" and intermingles with the atmosphere or rather the mass which encircles the earth. In all stages relating to the volume, weight or pressure of air, whether it is free or compressed, it will vary at all times with temperatures under specific conditions. If a volume of air is brought under pressure as in the confined state in a receiver, the air will increase in heat proportions, depending upon the space.

The opposite temperature may be created when, for instance, as in the use of refrigeration the volume of air is surrounded with a steady flowing stream of water; it may be brought to the freezing point.

This heat creation would counteract the intrinsic value of the use of air for industrial purposes, were it not for the fact that in cases where air being compressed to high pressure a method of cooling minimizes the heat and by water-cooling process the normal temperature is established. We term this stage-cooling and the expression is used when speaking of multi-stage compressors.

On the other hand, if cooling means are used, the work is less than is required for adiabatic compression, and the efficiency as compared with adiabatic compression therefore may approach or exceed unity, and this might conceivably be true even as compared with isothermal compression. Where cooling means are employed the compression curve ordinarily lies between the adiabatic and the isothermal lines and with good cooling is close to the isothermal.



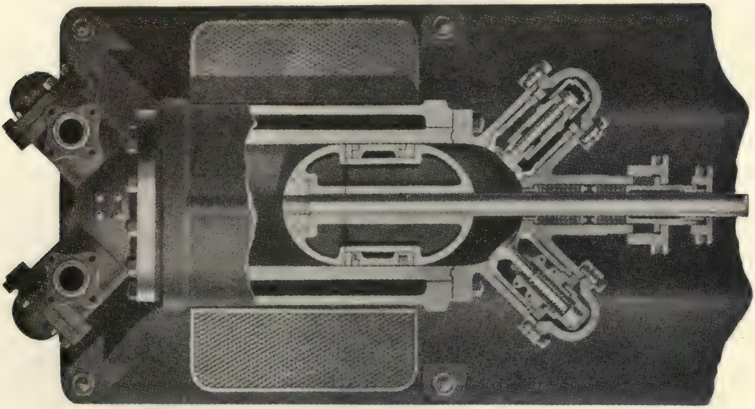
Compressor installation for Stationary Diesel Plant, Sullivan Type

The usual basis for the comparison of compressors of all types is the number of foot-pounds required at the shaft or horse power capacity of main engine to be required to produce a cubic foot of free air at a stated temperature compressed to a given pressure. The important consideration is the efficiency of the complete unit, including engine and compressor. In this respect, the efficiency performances of the engine, acting as the driving medium to cause the air to be compressed for the operating use on Diesels, is the factor upon which the performances of the compressor depends.

Cause of Defective Mechanism: The temperature due to compression depends upon three factors: (1) Initial temperature before compression; (2) Pressure to which air is compressed; (3) Efficiency of cooling

devices. No account will be taken of the effect of moisture in the air, and all temperatures given are for dry air.

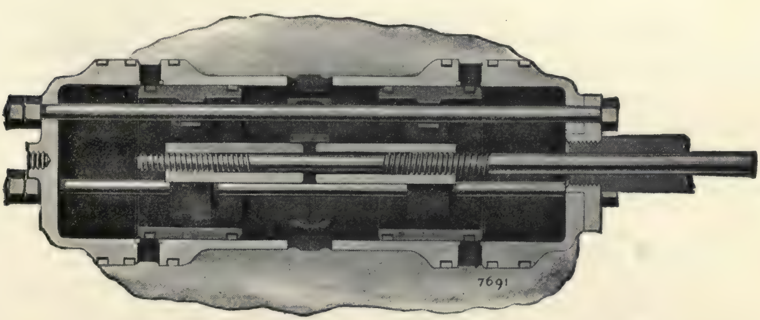
The place from which the air is drawn may have a very important bearing on initial temperature. The engine room is, to be sure, in the case of Diesel engine, the place where a compressor must be located. It is therefore to be expected that in this case the initial temperature is exceedingly high. A difference of 50 degrees between the engine room and



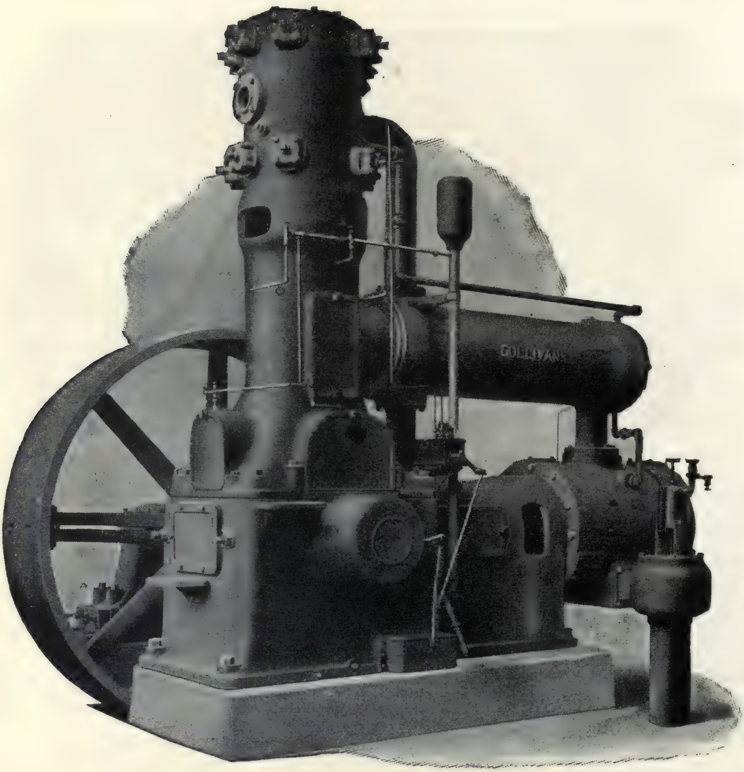
Cylinder Arrangement, Vilter Compressor

the outside air means more than a difference of 50 degrees in terminal temperature, as well as a loss of about 10 per cent in the capacity for the same amount of power expended.

The effect of leaking discharge valves upon initial temperature and consequently upon the temperature after compression, may be very serious indeed. Suppose an extreme case, where the amount of leakage is just sufficient to maintain atmospheric pressure within the cylinder, so that no fresh air enters.

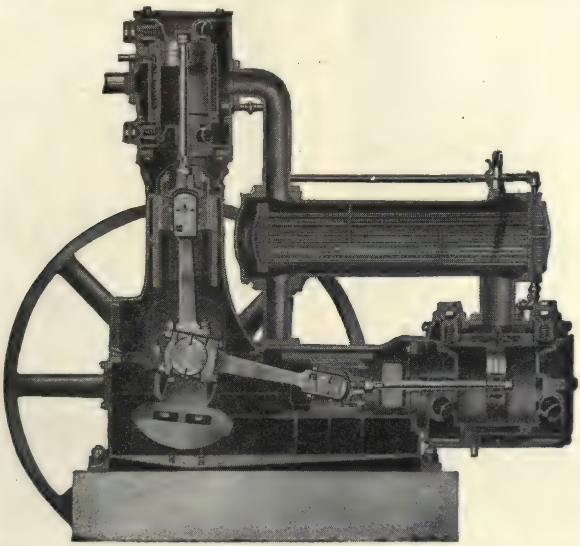


Stuffing Box and Piston Rod, Vilter Compressor



Sullivan W-J 3 Angle Compound Compressor, Full View

The initial temperature is now nearly the same as the terminal temperature of the previous charge, for the compressed air, in leaking back, has done no work upon the piston, and consequently has not dropped any in temperature. The hot air now receives a second compression and the terminal temperature, reached by starting from an initial temperature due to the previous stroke, may easily reach the point of ignition of combustible matter. If the initial temperature were 60 degrees F., the terminal pressure 40 pounds, the terminal temperature, with no cooling, 300 degrees F. If this air at 300 degrees F. leaks back and compression starts from that temperature, the temperature of discharge becomes 650 degrees F. With a discharge valve stuck open it is plain that in one stroke of the compressor a temperature might be reached sufficient to ignite the best grade of high flash cylinder oil.



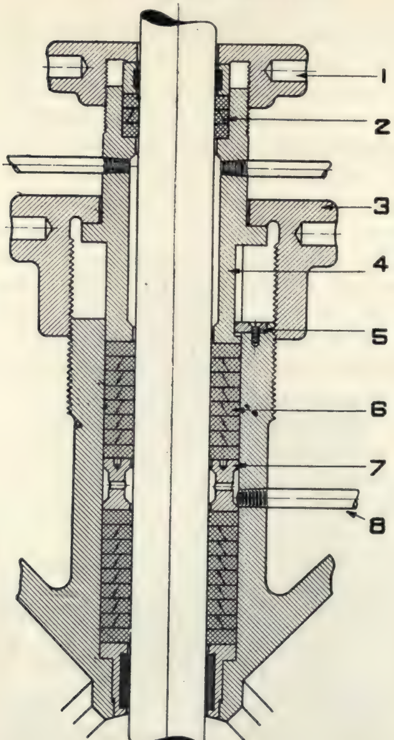
Cross Sectional View of Type "W-J 3" Angle Compound Compressor, Equipped with Inter-cooler

Operation of Compressor: (1) High flash test cylinder oil alone should be used for regular lubrication. Under no circumstances must kerosine or light oil be introduced. If an extra heavy dose of lubricant is required, give it soap and water through the oil pump.

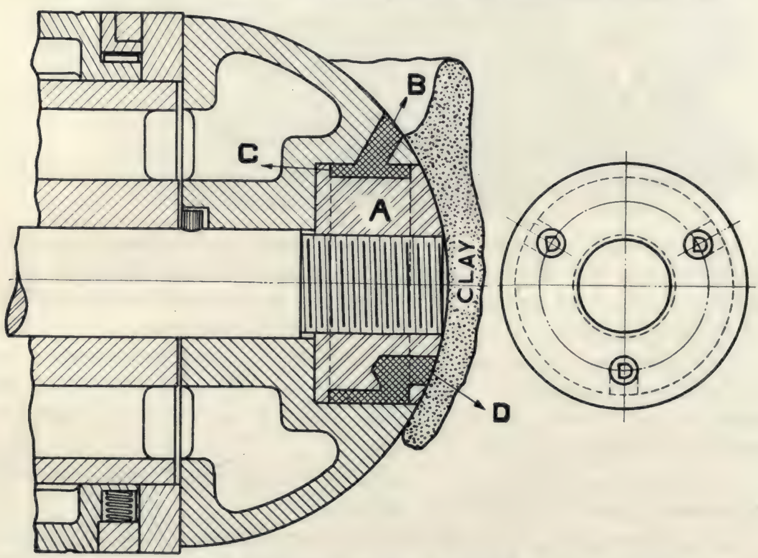
(2) Discharge valve must be kept tight, and to this end the use of an indicator is advised. The cards may not tell much about the conditions of the valves, but one of the greatest values of the indicator is the moral effect upon the engineer.

(3) Discharge valve must be cleaned from dust and oil and frequent examinations made to see if they need it.

(4) Accumulations of water and oil must be blown from the receiver and air starting bottles and an internal examination made at stated in-



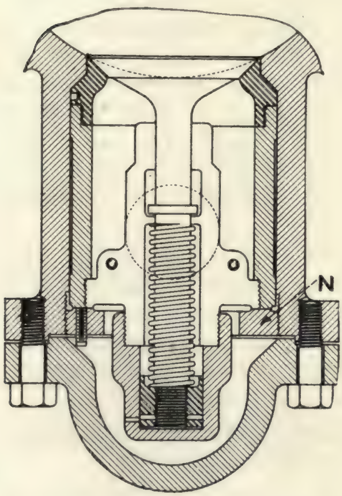
Cross-Sectional View of Stuffing Box of Vilter Air Compressor



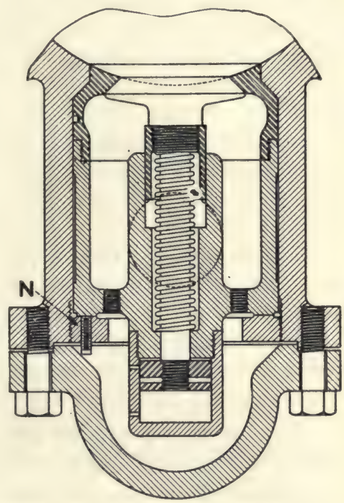
Sectional Plan of Cylinder Head on Vilter Air Compressor

tervals. The responsibility of operation of the engine depends upon the compressor, and that rests on the engineer in charge. He should be thoroughly instructed as to the possibility of explosion, the dangers attendant upon the use of any but prescribed oil, and the effect of leaking discharge valves or other mechanical defects. He should be acquainted with the use of indicator apparatus and required to submit cards at stated intervals. He should record in the engine room log the daily condition of the machine under his charge. He should be given a wholesome respect for an air compressor, with imperative knowledge as to the requirements of the same.

Knock in Cylinder: The clearance between the air pistons and the air cylinder heads is in most compressors about one-sixteenth of an inch. This clearance is carefully adjusted for proper functioning while in ser-



Suction Valve of Vilter Type of Compressor



Discharge Valve of Vilter Type of Compressor

vice, but in course of time, as wear takes place in the crankshaft bearings and connecting rod boxes, this clearance may be gradually reduced on one side until the piston strikes the head. This will be indicated by a pound as the crank passes the dead center on one end. When this is observed, shut down at once, remove a discharge valve and cage from each end of the cylinder, slack off the check nut on the piston rod where it screws into the crosshead, and screw the rod in or out until the clearance is even on both ends. This may be determined by bending an offset in a piece of soft steel or lead wire about one-sixteenth of an inch in diameter, and passing it through the discharge valve openings. When the compressor is turned over slowly the piston will compress the part of the wire projecting between it and the head, and show the amount of clearance.

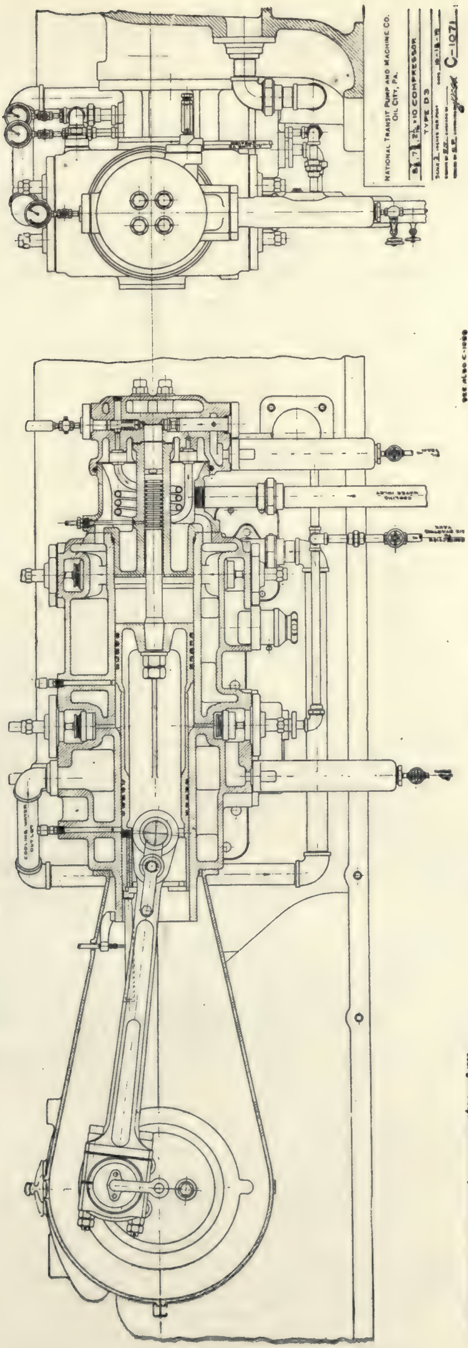
Inlet valves: The inlet valves should be removed occasionally for examination. To remove the inlet valves of most compressors, unscrew the plugs covering the valves. Usually the valves, together with its cage, will come out by giving a slight pull on the projecting end of the valve, as the cage has an easy sliding fit in the opening. If it does not pull out easily, it can ordinarily be started by working the valve up and down in the cage so as to produce a series of sharp blows on the cage. If the valve has not been removed for a long period, the cage sometimes becomes stuck in the opening, due to gumming of the cylinder oil, in the points. This dried oil may be readily softened by taking out a discharge valve at either end of the cylinder and pouring about a quart of kerosene into the cylinder on either side of the piston. This will work its way into the joints around the cages and allow them to be removed readily. If the cage still resists all efforts to remove it, it may be necesary, in order to remove the valve from the outside end of the cylinder, to take off the outside cylinder head and drive the valves out by means of a block of wood. To drive out the valves on the inside or frame end of the cylinder, it will be necessary to unscrew the piston rod from the crosshead and remove the piston from the cylinder. This last operation, however, is only necessary where the valves have been left in position for long periods and where the cylinder oil has been of extremely poor quality.

It is an excellent idea to have an extra valve, cage, spring and plug on hand. Then every week take out one valve and cage and replace it with the extra one. The valve and cage removed may then be examined, cleaned and refitted at leisure. The next week another valve may be removed and replaced; in this way all the valves will be regularly inspected and any wear or defect discovered before it becomes serious.

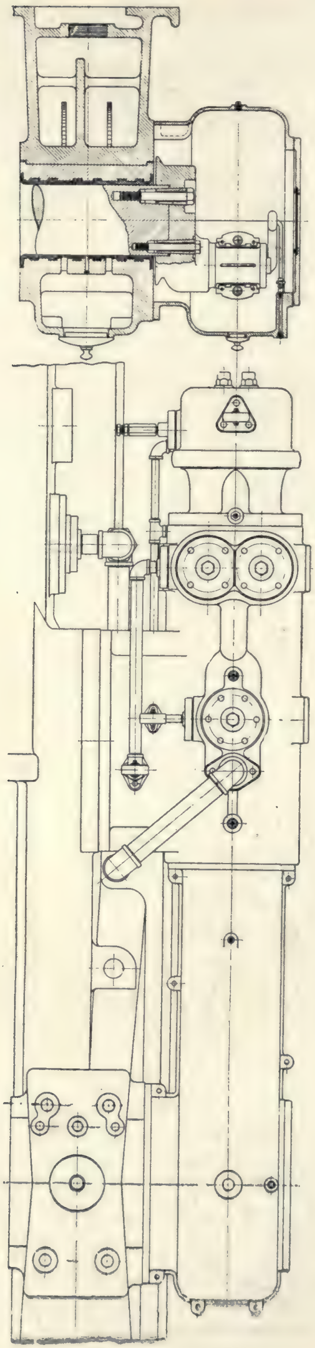
Thoroughly remove all oil from the valves and before replacing, smear the plugs and interior of the valves with air cylinder oil.

A mixture of graphite and cylinder oil placed on the threads of the plugs before putting them into place will allow them to be removed without difficulty at any time. Screw up the plugs firmly, so that the cages will not become loose and play back and forth.

Discharge Valves: In case a discharge valve needs regrinding, a special regrinding attachment should be on hand. The same may be obtained at small cost most anywhere. In most machines the discharge valves seat directly on the cylinder. Avoid leaky discharge valves. The principal troubles in lack of effiency on compressors are traceable to this defect. A leaky valve may be observed by the gauge and an escaping noise making a whistling sound. The inimical substances in air may cause corrosion. This defect will ultimately cause pitting of material with the consequential detrimental breakdowns.



Diagramatical View of National Transit Compressor

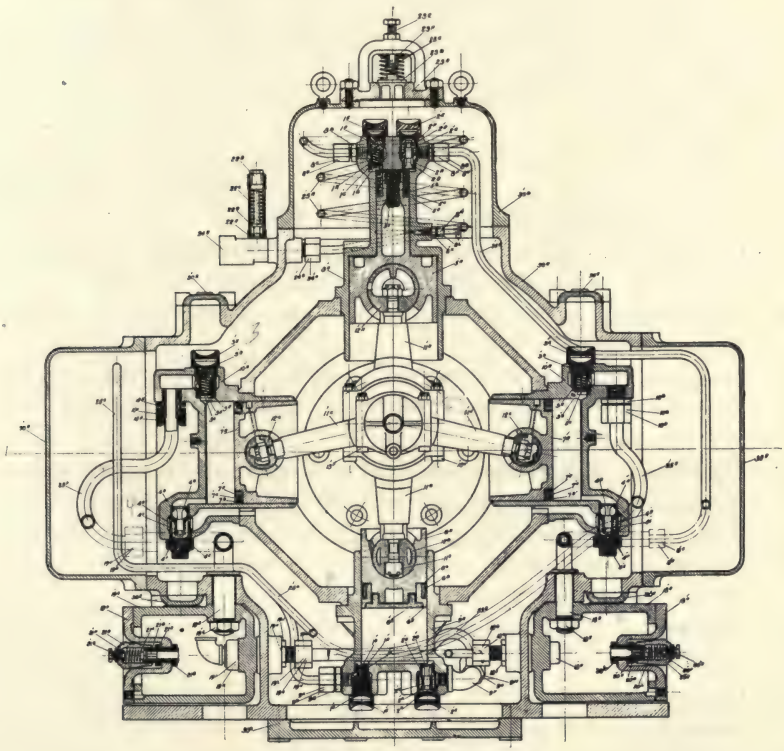


Plan View of Compressor of National Transit Oil Engine.

THE REAVELL THREE-STAGE REVERSIBLE AIR COMPRESSOR

Type Used on the Dow Diesel Engine

The Reavell patented Air Compressor, as shown in the illustration, is a three-stage reversible type and mounted on the end of the main engine bedplate, driven directly from an eccentric pin, which is secured to the main crankshaft. This compressor is of ample size to supply all the air required for starting and reversing purposes, as well as that required for injecting and atomizing the fuel oil.



Three-stage Reversible Reavell Air Compressor as used on the Dow Diesel Engine.

Definitions of Parts of Reavell Air Compressor:

1. **H.P. Discharge Valve.**—(A) Seat; (B) Valve; (C) Spring; (D) Plug; (E) Cap.
2. **H.P. Suction Valve.**—(A) Seat; (B) Valve; (C) Spring; (D) Plug; (E) Cap; (F) Collar.
3. **L.P. Discharge Valve.**—(A) Seat; (B) Valve; (C) Spring; (D) Plug; (E) Cap.
4. **L.P. Suction Valve.**—(A) Seat; (B) Valve; (C) Spring; (D) Plug; (E) Cap; (F) Collar.
5. **H.P. Piston.**—(A) Piston; (B) Outer Ring; (C) Inner Ring; (D) Carrier Ring; (E) Follower.
6. **I.P. Piston.**—(A) Piston; (B) Outer Ring; (C) Inner Ring; (D) Bull; (E) Follower.
7. **L.P. Piston.**—(A) Piston; (B) Outer Ring; (C) Inner Ring; (D) Follower.
8. **H.P. Cylinder.**—(A) Cylinder; (B) Discharge Fitting; (C) Union Nut; (D) Ring; (F) Suction Fitting; (G) Union Nut; (H) Ring; (K) Oil Fitting; (L) Union Nut; (M) Ring.
9. **I.P. Cylinder.**—(A) Cylinder; (B) Discharge Fitting; (C) Union Nut; (D) Ring; (F) Suction Fitting; (G) Union Nut; (H) Ring.
10. **L.P. Cylinder.**—(A) Cylinder; (B) Discharge Fitting; (C) Union Nut; (D) Ring.
11. **Connecting Rods.**—(A) H.P. Connecting Rod and Nut; (B) I.P. Connecting Rod and Special Nut; (C) L.P. Connecting Rod and Nut; (E) Special Nut.
12. **Gudgeons.**—(A) H.P. Gudgeon; (B) I.P. Gudgeon; (C) L.P. Gudgeon.
13. **Retainer Rings.**—(A) H. and I.P. Inner Retainer Ring; (B) H. and I.P. Outer Retainer Ring; (C) L.P. Inner Retainer Ring; (D) L.P. Outer Retainer Ring; (E) H. and I.P. Retainer Ring Bolts and Nuts; (F) L.P. Retainer Ring Bolts and Nuts.
14. **Bushings.**—(A) H.P. Bushing; (B) L.P. Bushing.
15. **Crank Pin Oiler.**
16. **Oiling Screw.**
17. **Crank Pin.**
18. **I.P. Purge Pot.**—(A) Body; (B) H.P. Suction Fitting; (C) Union Nut; (D) Ring; (E) Nut and Washer; (F) I.P. Discharge Fitting; (G) Union Nut; (H) Ring; (J) Nut and Washer; (O) Cover.
19. **L.P. Purge Pot.**—(A) Body; (B) L.P. Suction Fitting; (C) Union Nut; (D) Ring; (E) Nut and Washer; (F) L.P. Discharge Fitting; (G) Union Nut; (H) Ring; (J) Nut and Washer; (K) I.P. Suction Fitting; (L) Union Nut; (M) Ring; (N) Nut and Washer; (O) Cover.

20. **I.P. Relief Valve.**—(A) Seat; (B) Valve; (C) Spring; (D) Body; (E) Screw; (F) Collar; (G) Nut.

21. **L.P. Relief Valve.**—(A) Seat; (B) Valve; (C) Spring; (D) Body; (E) Screw; (F) Collar; (G) Nut.

22. **H.P. Relief Valve.**—(A) Seat; (B) Valve; (C) Spring; (D) Screw.

23. **Water Relief Valve.**—(A) Seat; (B) Valve; (C) Inner and Outer Springs; (E) Screw; (F) Collar.

24. **Final Delivery Fitting.**—(A) Body; (B) Union Nut; (C) Ring.

25. **Air Piping.**—(A) H.P. Discharge Pipe; (B) H.P. Suction Pipe; (C) I.P. Discharge Pipe; (D) I.P. Suction Pipe; (E) Long L.P. Discharge Pipe; (F) Short L.P. Discharge Pipe.

26. **Lubricator Fittings.**—(A) Lubricator; (B) Fitting; (C) Union Nut; (D) Ring; (E) Nut and Washer.

27. **Oil Drip Fittings.**—(A) Oil Drip; (B) Fitting; (C) Union Nut; (D) Ring.

28. **Leading Pipes.**

29. **Connection Parts.**

30. **Casing.**—(A) Main Casing; (B) H.P. Cover; (C) I.P. Cover; (D) L.P. Cover; (E) Inspection Cover; (F) Front Cover; (G) Name Plate Cover.

EFFLUX OF AIR

As the pressure is dependent upon both the height and the density of the fluid, it is evident that for a given pressure, the less the density the greater the height of the column. But the law of fallen bodies recognizes the fact that it is the distance fallen through and not the weight of the body that determines the velocity. Therefore, the less dense a body the higher the column required to produce a given pressure and the greater the velocity of discharge. From this it is evident that the velocity of a gas issuing under a given pressure would be greater than that of a liquid under the same conditions. And conversely, the more dense the fluid issuing at a given velocity the greater must have been the pressure to produce that velocity.

In the case of a liquid, the atmospheric pressure upon the inlet and outlet of a containing vessel is balanced and the actual height or head may be actually measured. But air is invisable, and there is no tangible distinction in substance between that producing the pressure and that constituting the surrounding atmosphere.

The pressure of the atmosphere is due to the weight of air, and, for any area, is to be measured by the weight of a column of air having the given area as a base and a height equal to that of the atmosphere. But this height cannot be accurately determined, and, furthermore the density of the air decreases in geometric ratio as the distance from the earth increases. For the purpose of calculation, however, the practical equivalent of such a column may be determined by assuming the air to be of uniform density throughout and the column of such a weight the same and to produce the same effective pressure per unit of area.

Under the standard conditions of barometric pressure of 29.921 inches, the atmospheric pressure is 14.69 pounds per square inch, or 2,115.36 pounds per square foot.

At this pressure a cubic foot of dry air at 50 degrees has a density of 0.077884 pounds. Consequently a homogeneous column

$$\frac{2{,}115.36}{0.077884} = 27{,}160 \text{ pounds}$$

and exerts this pressure upon the given area.

EFFICIENCY OF COMPRESSOR

The efficiency of a reciprocating air compressor is ordinarily stated as the ratio of the work done upon the air. If air be compressed in a non-conducting vessel without commotion or friction, its temperature will rise in a definite and known way as its pressure is increased and its volume diminished. Such compression is known as adiabatic compression, since heat does not leave nor center the air as heat during the

process. On the other hand, if the air be slowly compressed and heat be withdrawn constantly during compression, so that the temperature is constant throughout the process, the final volume when compressed to a stated pressure is less than in adiabatic compression, or if compressed to a given volume, the pressure will be less, as likewise, in either case, will be the work of compression. This is called isothermal compression. If the air is cooled so much that the final temperature is lower than the initial temperature, the pressure or volume, or both, and the work will be reduced below that corresponding to isothermal compression. The standard of isothermal compression, while therefore in a sense arbitrarily, may properly be used when stating the efficiency of compressors which are equipped with cooling means or intercoolers, as it gives a measure of the combined efficiency of the cooling means and of the means of compression.

In a "perfect" uncooled compressor the relation between pressure and volume would follow the adiabatic law, which therefore supplies a rational standard for comparison. In actual compressors not supplied with cooling means, the volume to which the air is reduced at the given pressure, or the pressure reached at a given volume reduction is always greater than that corresponding to adiabatic compression, as is also the amount of work required. In actual compressors where cooling means are not employed, the final temperature is always higher than that which would be obtained in adiabatic compression, due to heat generated by friction of the air. As compared with adiabatic compression, compressors without cooling means may reach efficiencies of 70 per cent. or above.

A factor which enters into the efficiency rating of high-pressure compressors is the usual basis for the comparison of all types in the number of foot-pounds required at the shaft by the engine per cubic foot of free air at a stated temperature compressed to a given pressure. The important consideration is the efficiency of the complete unit, including motor and compressor.

Simultaneously with the variations in head and delivery at constant speed, there are corresponding variations in the power consumed and in the efficiency necessary to supply the amount of air in Diesel engineering.

Variation of Dry Air According to Temperature: The volume of one pound of dry air at 32 degrees Fahrenheit and 30-inch barometer is 12.38 cubic feet. This volume varies directly as the absolute temperature, and therefore, for a temperature of 90 degrees Fahrenheit the volume is

$$12.38 \times \frac{460 + 90}{460 + 32} = 13.84 \text{ cubic feet.}$$

WEIGHT OF DRY AIR IN POUNDS PER CUBIC FOOT

30-IN. BAROMETER.

°F.	Lbs. Cu. Ft.	°F.	Lbs. Cu. Ft.	°F.	Lbs. Cu. Ft.
0	.0863	210	.0593	600	.0374
10	.0845	220	.0584	650	.0358
20	.0827	230	.0575	700	.0342
30	.0810	240	.0566	750	.0328
40	.0794	250	.0559	800	.0315
50	.0779	260	.0551	850	.0303
60	.0764	270	.0544	900	.0292
70	.0749	280	.0536	950	.0282
80	.0736	290	.0529	1000	.0272
90	.0722	300	.0522	1100	.0254
100	.0709	320	.0509	1200	.0239
110	.0696	340	.0496	1300	.0226
220	.0685	360	.0484	1400	.0214
130	.0673	380	.0473	1500	.0203
140	.0661	400	.0461	1600	.0913
150	.0651	420	.0451	1700	.0184
160	.0640	440	.0441	1800	.0176
170	.0630	460	.0432	1900	.0168
180	.0620	480	.0422	2000	.0161
190	.0611	500	.0414	3000	.0115
200	.0602	550	.0393		

Centrifugal Compressor Formula: In general the pressure generated per stage by a centrifugal blower or compressor may be represented by following formula:

$$P = C \frac{U^2}{g} D$$

wherein D is the density, C is an arbitrary co-efficient, U is the rim velocity of the impeller in feet per second, and g is the acceleration of gravity, equal to 32.2 in the English system of units. The limiting value of U is controlled by the speed of revolution, which may be imposed by the driving motor, the strength of materials available and the volume handled. The value of C depends upon the shape of the blades and the efficiency of the diffusor and in blowers and compressors suitable for Diesel operation has a value in the neighborhood of 0.5.

THEORETICAL LEAKAGE OF AIR AT 70 DEGREES FAHRENHEIT

Effective Draft in Inches of Water	Leakage in pounds per hour per square in. of opening
0.2	56
0.4	79
0.6	97
0.8	112
1.0	125
1.5	153
2.0	177
2.5	197
3.0	216
3.5	234

The tabular values above are for the ideal case of zero friction and contraction and must be multiplied by a co-efficient C, to obtain the actual leakage. For the equivalent of an orifice in a thin plate, C = 0.6 approximately. For a short cylindrical pipe with inner corners not rounded, C = 0.75 approximately.

MEAN BAROMETER PRESSURES CORRESPONDING TO ALTITUDES FROM 100 TO 4900 FT. ABOVE SEA LEVEL

Altitude Feet	Pressure Inches	Altitude Feet	Pressure Inches	Altitude Feet	Pressure Inches
0	30.00	1700	28.14	3400	26.35
100	29.88	1800	28.04	3500	26.26
200	29.76	1900	27.94	3600	26.16
300	29.64	2000	27.82	3700	26.06
400	29.52	2100	27.70	3800	25.96
500	29.40	2200	27.58	3900	25.85
600	29.29	2300	27.47	4000	25.75
700	29.18	2400	27.36	4100	25.64
800	29.08	2500	27.26	4200	25.55
900	29.97	2600	27.17	4300	25.46
1000	28.86	2700	27.05	4400	25.37
1100	28.76	2800	26.95	4500	25.26
1200	28.65	2900	26.85	4600	25.16
1300	28.54	3000	26.74	4700	25.07
1400	28.44	3100	26.65	4800	24.98
1500	28.34	3200	26.55	4900	24.88
1600	28.23	3300	26.45		

RELATIVE HUMIDITY TABLE

BAROMETER 30″

DIFFERENCE BETWEEN DRY AND WET THERMOMETERS

AIR TEMPERATURES	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	AIR TEMPERATURES
30	100	89	78	67	57	47	36	26	17	7																				30
35	100	91	82	73	65	54	45	37	28	19	12	3																		35
40	100	92	84	76	68	60	53	45	38	30	22	16	8	1																40
45	100	92	85	78	71	64	58	51	44	38	32	25	19	13	7	1														45
50	100	93	87	80	74	67	61	55	50	44	38	33	27	22	16	11	6	1												50
55	100	94	88	82	76	70	65	59	54	49	43	39	34	29	24	19	16	10	6	1										55
60	100	94	89	84	78	73	68	63	58	53	48	44	39	34	30	26	22	18	14	10	6	2								60
65	100	95	90	85	80	75	70	65	61	56	52	48	44	39	35	31	28	24	20	17	13	10	6	3						65
70	100	95	90	86	81	77	72	68	64	60	55	52	48	44	40	36	33	29	26	23	19	16	13	10	7	4	1			70
75	100	95	91	87	82	78	74	70	66	62	58	55	51	47	44	40	37	34	31	27	24	21	19	16	13	10	7	5	2	75
80	100	96	92	87	83	79	75	72	68	64	61	57	54	51	47	44	41	38	35	32	29	26	23	20	18	15	13	10	8	80
85	100	96	92	88	84	80	77	73	70	66	63	60	56	53	50	47	44	41	38	36	33	30	28	25	22	20	17	15	13	85
90	100	96	92	88	85	81	78	75	71	68	65	62	59	56	53	50	47	44	41	39	36	34	32	29	26	24	22	20	17	90
95	100	96	93	89	86	82	79	76	72	69	66	63	60	58	55	52	49	47	44	42	39	37	35	32	30	28	25	23	21	95
100	100	97	93	90	86	83	80	77	74	71	68	65	62	59	57	54	51	49	47	44	42	39	37	35	33	31	29	27	25	100
105	100	97	93	90	87	84	81	78	75	72	69	66	64	61	58	56	53	51	49	46	44	42	40	38	35	33	31	30	28	105
110	100	97	94	90	87	84	81	78	76	73	70	67	65	62	60	57	55	53	50	48	46	44	42	40	38	36	34	32	30	110
	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	

CHAPTER XIV.

PUMPS

SOME FACTS ON PUMPS

Pumps may be defined in two classes; namely, the Plunger Pump, working upon the plunger system, and the Centrifugal Pump, which operates by rotary or centrifugal motion. The first named type may be termed "pressure" pump of either the low pressure or high pressure type, depending upon the construction of the same. Where the necessity of pumping masses is called for, the centrifugal pump answers this purpose.

The centrifugal pump is the most suitable of all the forms of pumps. Its simplicity in construction and the adaptability to be operated by electric power creates a greater demand for this machine.

It can be coupled direct to the armature shaft of the motor either with a rigid connection or an elastic coupling, preferably the latter, and may be driven at any speed from 500 revolutions per minute up to 2000 revolutions per minute.

It has one conspicuous advantage over the three-throw and similar forms of pumps, in that it has no valves, and is in consequence able to pump muddy or gritty water without damaging the working parts or hindering its action in any way.

The centrifugal pump is thus particularly suitable for disposing of the discharge water from coal-washing machines, or for use in construction work, providing the head of the water is not too great. The centrifugal pump in its simplest form consists of a number of curved blades arranged round a central axle or shaft, and revolving in an approximately circular casing which is connected up to the delivery pipe or column.

Both in outward appearance and internal construction the centrifugal pump is, therefore, not unlike the ordinary centrifugal fan.

Its action, too, depends upon the same principle, namely, centrifugal force. The water contained between the blades of the pump, by reason of the centrifugal force, is thrown off at a tangent, and finds escape at the orifice leading to the discharge pipe or column.

Until quite recently the main objection to the centrifugal pump has been the very low efficiency obtained, and as the limit of working head of a single pump is about 70 feet, this has entailed the use of the cumbersome combinations for higher lifts. These objections, however, cannot now be urged against the centrifugal pump, as by coupling up two or more single pumps in series it is possible to throw water to any height up to 1000 feet, and still obtain a very good efficiency.

The principle feature in the multiple chamber centrifugal pump is that it consists of one or more sets of vanes or impellers, each running in its own chamber, but upon a common shaft, the delivery pressure of the liquid varying directly as the number of chambers is used. Thus, if an ordinary single pump can deliver water against a head of 70 feet, the addition of another chamber will give a final delivery head of 140 feet, while four chambers will enable the pump to discharge the same amount of water against a total head of 280 feet.

Of late a great deal of use is made with the Turbine Pump. In this type the water enters the revolving wheel axially, traverses the curved internal passages between the vanes, and is discharged tangentially at the periphery into a stationary guide ring; this conveys it to the annual chamber in the body of the pump, where the velocity head imparted to the water by the wheel is converted into pressure head.

From this chamber the water is finally discharged into the pipe lines, or, if the pump may be a multiple one, into the second and subsequent chambers. A special feature of this pump is the provision of the stationary guide ring mentioned above; this is fixed concentric with the revolving vanes, and, owing to its design, enables the conversion of the velocity into pressure head to be very effectively accomplished, thus increasing not only the possible height of lift, but also the working efficiency of the pump. The ideal source of power for working centrifugal and turbine pumps is undoubtedly the direct coupled electric motor.

The Turbine Pump possesses many advantages, conspicuous amongst these being the small number of working parts, compactness, low first cost, and minimum of wear and tear.

In calculations relating to the centrifugal and turbine pumps the following formula will be helpful:

Let S = speed of periphery of wheel in feet per second.

Let H = height in feet to which water is to be delivered.

Let D = diameter of wheel in feet.

Let G = gallons of water delivered per minute.

Let R = revolutions per minute.

The horsepower of motor required will be found by multiplying the height in feet by the quantity of water in pounds delivered per minute, and by the efficiency of the pump and motor, and dividing by 33,000. The efficiency of the pump may be anything from 0.55 to 0.65, and the efficency of the motor, say, 0.85, the combined efficiencies being thus equal to from 70 to 75 per cent.

The average slippage of a pump is about 20 per cent.

Suppose we had a pump that had the actual displacement of 130 gallons per minute, and it only pumped 100 gallons per minute, how would we find the actual slippage?

The percentage will be 100 gallons divided by 13 = .7461 × 100 = 74.61%. Deduct this from 100 per cent. will equal 25.39% slippage.

Table Showing Capacities of Pumps in U. S. Gallons

Diam. Pumps in Inches	Piston Speed in Feet per Minute											Diam. Pumps in Inches
	40	50	60	70	80	90	100	125	150	175	200	
1½	3.67	4.58	5.51	6.42	7.34	8.25	9.17	....	....	....	....	1½
1¾	5.00	6.25	7.49	8.75	10.00	11.25	12.50	....	....	....	....	1¾
2	6.53	8.15	9.79	11.41	13.06	14.67	16.32	....	....	....	....	2
2¼	8.26	10.32	12.39	14.45	16.52	18.58	20.65	....	....	....	....	2¼
2½	10.20	12.75	15.30	17.85	20.40	22.95	25.50	....	....	....	....	2½
2¾	12.34	15.42	18.51	21.59	24.68	27.67	30.85	....	....	....	....	2¾
3	14.69	18.36	22.03	25.70	29.38	33.04	36.72	....	....	....	....	3
3¼	17.24	21.54	25.86	30.16	34.48	38.78	43.09	....	....	....	....	3¼
3½	19.99	24.99	29.99	34.98	39.98	44.98	49.98	62.47	....	....	....	3½
3¾	22.95	28.68	34.42	40.15	45.90	51.63	57.37	71.72	....	....	....	3¾
4	26.11	32.64	39.17	45.19	52.22	58.75	65.28	81.60	97.92	....	....	4
4¼	29.48	36.84	44.22	51.58	58.96	66.32	73.69	92.12	110.54	....	....	4¼
4½	33.05	41.31	49.57	57.83	66.10	75.35	82.62	103.27	123.93	144.59	....	4½
4¾	36.82	46.02	55.23	64.43	73.64	82.84	92.05	115.07	138.08	161.10	....	4¾
5	40.80	51.00	61.20	71.00	81.60	91.80	102.00	127.50	153.00	178.50	204.00	5
5¼	44.78	56 22	67.47	78.71	89.56	101.20	112.45	140.51	168.68	196.67	224.91	5¼
5½	49.37	61.70	74.05	86.39	98.74	111.07	123.42	154.27	185.13	215.98	246.84	5½
5¾	53.96	67.44	80.94	94 42	107.92	121.40	134.89	168.62	202.34	236.07	269.79	5¾
6	58.75	73.44	88.13	102.71	117.50	132.19	146.88	183.60	220.32	257.04	293.76	6
6½	68.95	86.19	103.43	120.66	137.90	155.14	172.38	215.47	258.57	301.66	344.76	6½
7	79.97	99.96	119.95	139.94	159.94	179.92	199.92	249.90	299.88	349.86	399.84	7
7½	91.80	114.75	137.70	160.65	183.60	206.55	229.50	286.88	344.25	401.62	459.00	7½
8	104.45	130.56	156.67	182.78	208.90	235.00	261.12	326.40	391.68	456.96	522.24	8
8½	117.91	147.39	176.87	206.34	235.82	265.30	294.78	368.47	442.17	515.86	589.56	8½
9	132.19	165.24	198.29	231.33	264.38	297.43	330.48	413.10	495.72	578.34	660.96	9
9½	147.29	184.11	220.93	257.75	294.58	331.39	368.22	460.27	552.33	644.38	736.44	9½
10	163.20	204.00	244.80	285.60	326.40	367.20	408.00	510.00	612.00	714.00	816.00	10
10½	179.93	224.91	269.89	314.87	329.86	404.83	449.82	562.27	674.73	787.18	899.64	10½
11	197.47	246.84	296.21	345.57	394.94	444.31	493.68	617.10	740.52	863.94	987.36	11
12	235.00	293.75	352.50	411.25	470.00	528.75	587.50	734.40	881.30	1028.20	1175.00	12
13	275.80	344.75	413.70	482.65	551.60	620.55	689.50	861.90	1034.30	1206.70	1379.00	13
14	319.90	399.85	479.80	559.79	639.70	719.73	799.70	999.60	1199.50	1399.40	1599.40	14
15	367.20	459.00	550.80	642.60	734.40	826.20	918.00	1147.50	1377.00	1606.50	1836.00	15
16	417.80	522.25	626.70	731.15	835.60	940.05	1044.50	1305.60	1566.70	1827.80	2089.00	16
18	528.80	660.95	793.20	925.33	1057.50	1189.71	1321.90	1652.40	1982.90	2313.40	2643.80	18
20	652.80	816.00	979.20	1142.42	1305.60	1468.80	1632.00	2040.00	2448.00	2856.00	3264.00	20
22	789.90	987.35	1184.80	1382.29	1579.80	1777.23	1974.70	2468.40	2962.10	3455.80	3949.40	22
24	940.00	1175.05	1410.00	1645.07	1880.10	2115.09	2350.10	2937.60	3525.10	4112.60	4700.10	24
26	1103.20	1379.05	1654.80	1930.67	2206.50	2482.29	2758.10	3447.60	4137.10	4826.60	5516.10	26
28	1279.50	1599.35	1919.20	2239.09	2559.00	2878.83	3198.70	3998.40	4798.10	5597.70	6397.40	28
30	1468.80	1836.00	2203.20	2570.40	2937.60	3304.80	3672.00	4590.00	5508.00	6426.00	7344.00	30
32	1671.20	2088.95	2506.70	2924.53	3342.30	3760.11	4177.90	5222.40	6266.90	7311.40	8355.80	32
36	2115.10	2643.85	3172.60	3701.39	4230.14	4758.93	5287.70	6609.60	7931.50	9253.40	10575.30	36
40	2611.20	3264.00	3916.80	4569.60	5222.40	6775.20	6528.00	8160.00	9792.00	11424.00	13056.00	40
48	3760.10	4700.15	5640.20	6580.21	7520.20	8460.27	9400.30	11750.40	14100.40	16450.50	18800.60	48

MEAN EFFECTIVE PRESSURE AND HORSEPOWER

Developed in Compressing a Cubic Foot of Free Air (Adiabatically) from Atm. Press. (14.7 lbs.) to Various Gauge Pressures. Initial Temp. of Air in Each Cylinder taken as 60° Fahn. Jacket Cooling not considered.

			Single Compression				Two-Stage Compression				
Gage Press-Pounds	Absolute Pressure Pounds	Number of Atmospheres Compressed	M. E. P Per Sq. Inch Theoretical	Developed H. P Theoretical	M. E. P. per Sq. In. 15 Percent Friction Included	Developed H. P Fifteen Percent Friction Included	M. E. P. per Sq. In. Reduced to L. P. Air Cyl. (Theor.)	Developed H. P. Theoretical	M. E. P. per Sq. In. Red. to L. P. Air Cyl., 15 Percent Friction Included	Developed H. P. 15 Percent Friction Included	Percent of Power Saved by Two-Stage over Single Compression (Theor.)
5	19.7	1.34	4.46	.019	5.12	.022					
10	24.7	1.68	8.21	.036	9.44	.041					
15	29.7	2.02	11.46	.050	13.17	.057					
20	34.7	2.36	14.30	.062	16.44	.071					
25	39.7	2.70	16.94	.074	19.47	.085					
30	44.7	3.04	19.32	.084	22.21	.096					
35	49.7	3.38	21.50	.094	24.72	.108					
40	54.7	3.72	23.53	.103	27.05	.118					
45	59.7	4.06	25.40	.111	29.21	.127					
50	64.7	4.40	27.23	.119	31.31	.136					
55	69.7	4.74	28.90	.126	33.23	.145					
60	74.7	5.08	30.53	.133	35.10	.153					
65	79.7	5.42	32.10	.140	36.91	.161					
70	84.7	5.76	33.57	.146	38.59	.168	29.31	.128	33.71	.147	12.7
75	89.7	6.10	35.00	.153	40.25	.175	30.43	.133	34.99	.153	13.0
80	94.7	6.44	36.36	.159	41.80	.182	31.44	.137	36.15	.158	13.5
85	99.7	6.78	37.63	.164	43.27	.189	32.46	.142	37.32	.163	13.8
90	104.7	7.12	38.89	.169	44.71	.195	33.37	.145	38.36	.167	14.2
95	109.7	7.46	40.11	.175	46.12	.201	34.28	.149	39.41	.172	14.5
100	114.7	7.80	41.28	.180	47.46	.207	35.20	.153	40.48	.176	14.7
110	124.7	8.48	43.56	.190	50.09	.218	36.82	.161	42.34	.185	15.4
120	134.7	9.16	45.69	.199	52.53	.229	38.44	.168	44.20	.193	15.9
130	144.7	9.84	47.72	.208	54.87	.239	39.86	.174	45.83	.200	16.5
140	154.7	10.52	49.64	.216	57.08	.249	41.28	.180	47.46	.207	16.9
150	164.7	11.20	51.47	.224	59.18	.258	42.60	.186	48.99	.214	17.2
160	174.7	11.88					43.82	.191	50.39	.219	
170	184.7	12.56					44.93	.196	51.66	.225	
180	194.7	13.24					46.05	.201	52.95	.231	
190	204.7	13.92					47.16	.206	54.22	.236	
200	214.7	14.60					48.18	.210	55.39	.241	
250	264.7	18.00					52.84	.230	60.76	.264	
300	314.7	21.40					56.70	.247	65.20	.283	
350	364.7	24.81					60.15	.262	69.16	.301	
400	414.7	28.21					63.19	.276	72.65	.317	
450	464.7	31.61					65.93	.287	75.81	.329	
500	514.7	35.01					68.46	.298	78.72	.342	
550	564.7	38.41					70.70	.308	81.30	.354	
600	614.7	41.81					72.83	.317	83.75	.364	

STROKES REQUIRED TO REACH A PISTON SPEED OF 100 Ft. PER MINUTE

Length of Stroke	Number of Strokes	Length of Stroke	Number of Strokes	Length of Stroke	Number of Strokes
4	300	12	100	24	50
5	240	14	86	26	46
6	200	16	75	28	43
7	172	18	67	30	40
8	150	20	60	36	33
10	120	22	55	40	30

HOW TO DETERMINE THE POWER REQUIRED FOR PUMPING

The power required for pumping depends primarily upon two factors—the weight of the liquid to be pumped per minute and the vertical height it has to be raised from the source of supply to the point of delivery. In addition to these two principal factors, allowance must be made in practice for the power required to overcome the losses in the pumping equipment and the friction in the pipe lines.

To get the power required in terms of horsepower: multiply the weight of the liquid to be pumped per minute in pounds, by the height it has to be lifted and forced in feet; and divide this by 33,000.

This gives the following formula:

$$\text{Horsepower} = \frac{\text{Wt. of Liquid per Min. in Lbs.} \times \text{Height pumped in Ft.}}{33{,}000}$$

The above gives the theoretical power required. The actual power needed to do the work will be in excess of this, the amount depending upon the losses. To get the actual power required, the figure representing the height pumped should be increased by the loss of head in feet due to friction in the pipe line.

The result determined in this way must then be corrected for the power loss in the pumping equipment. This is accomplished by dividing the horsepower obtained, by the efficiency of the pumping outfit, expressed in a decimal; with these corrections our formula becomes:

$$\text{H. P.} = \frac{W \times H}{33{,}000 \times E}$$

Where W is the weight of liquid pumped per minute in pounds, H is the total head in feet including loss of head due to friction in pipe lines and E is the efficiency of the pump.

Having determined the horsepower required, the cost of operating the pump per hour can be obtained by multiplying this figure by the cost per horsepower-hour of operating the engine or motor used to drive the pump.

Another formula commonly used in practice for determining the power required for pumping water is:

$$\text{H. P.} = \frac{\text{G.P.M.} \times \text{H}}{3{,}960 \times \text{E}}$$

Where G.P.M. is the gallons of water per minute, H is the total head in feet including loss of head due to friction in pipe, and E is the efficiency of the pump, expressed as a decimal. This formula is satisfactory for all practical purposes, where water is the liquid pumped.

The actual brake horsepower required to drive a centrifugal pump can be found by the following formula:

$$.000253 = \frac{\text{U. S. Gallons per Minute} \times \text{Head in Feet}}{\text{Pump Efficiency Per Cent}} = \text{Brake Horsepower}$$

In pumping fuel oil, where the liquid being heavier than water it being necessary, to multiply by the specific gravity.

The horsepower of the motor selected should be approximately ten per cent greater than the brake horsepower as centrifugal pumps are designed to be slightly over capacity.

To figure current consumption per thousand gallons of water pumped the following formula will be found convenient:

$$\frac{\text{Head in feet} \times .0031456}{\text{Pump Efficiency} \times \text{Motor Efficiency}} = \text{KWH's per 1000 gallons pumped.}$$

To figure duty for Diesel driven pumps the following formula can be used:

$$\frac{1980 \times \text{Pump Efficiency}}{\text{Constant pressure per B.H.P. per hour}} =$$

Foot Pounds duty per Lbs. Constant Pressure of Engine.

INFORMATION ON PUMPS

(Constants and Formulas)

V = Velocity feet per second.

D = Diameter pipe in inches.

A = Area square inches.

Q = Water quantity, G.P.M.

$$V = \left[\frac{2.02}{D} \right]^2 \times \frac{Q}{10} \times \frac{3.21}{A} = \frac{Q}{10}$$

$$Q = 2.45\ VD^2$$

$$\text{Gal. per 24 hours} = 3530 \times V \times D^2$$

$$Q = \frac{10 \times A \times V}{3.21}$$

$$\text{Velocity head, } h = \frac{V^2}{2g} = 32.16 \text{ ft. per sec.}$$

Pump Formula: In determining the size of a pump required to deliver a given number of gallons through pipes, allowance must be made for friction in the pipes. This places additional work on the pump and is figured as so much additional head, which is called friction head.

If H = loss of head due to friction, or friction head in feet,
L = length of straight pipe in feet,
V = velocity of flow in feet per second,
D = diameter of pipe in feet;

we have the following formula for Friction Head:

$$H = \frac{.02LV^2}{64.\ 4D}$$

Pump Horse Power Formula: If allowance is made for the friction of the flow in the pipe, we have the following:

$$\text{H.P.} = \frac{\text{Lbs. water per minute } (H + 2.31\ P + H)}{33{,}000}$$

In which H.P. = pump horsepower required,
H = suction lift in feet,
P = pressure developed in pipe in lbs. per sq. in.
which = pressure delivered by pump minus pressure at which water is delivered to pump,
H = friction head allowance in feet.

USEFUL INFORMATION.

1 cubic foot of water	62.3791 lbs.
1 cubic inch of water	.03612 lbs.
1 gallon of water	8.338 lbs.
1 gallon of water	231. cubic ins.
1 cubic foot of water	7.476 gallons
1 pound of water	27.7 cubic inches

A Gallon of Water (United States Standard) weighs 8 1/3 pounds, and contains 231 cubic inches. A cubic foot of water weighs 62½ pounds and contains 1.728 cubic inches, or 7½ gallons.

On pump formula use this:

P = pressure per lbs. per sq. in.	$P = H \times .4335$
H = head of water in feet	$H = P \times 2.307$
Pressure per square foot =	$H \times 62.425$

Pressures: 1 atm. = 14.7 lbs. sq. in. at sea level. Roughly the barometric pressure decreases ½ lb. per sq. in. per 1000 ft. ascent.

1 lb. per sq. in. = 2.0416 in. Mercury at 62° F.
= 2.0355 in. Mercury at 32° F.
= 27.71 in. Water at 62° F.
= 2.309 ft. of Water at 62° F.
= 0.0703 kg. per sq. cm.
1 ft. of Water at 62° F. = .433 lb. per sq. in.
1 in. of Mercury at 62° F. = .491 lb. per sq. in.
= 1.132 ft. of Water at 62° F.

The above data is calculated for distilled water at 40 degrees Fahrenheit.

To compute the horsepower necessary to raise water to any height: Multiply the gallons per minute by 8.33. Multiply this product by the feet lift. The result is foot pounds. Divide by 33,000 to reduce to horsepower per minute. (An allowance of 25 per cent should be made for friction, etc.)

To compute the capacity of pumping engines, multiply the area of the water piston, in inches, by the distance it travels, in inches, in a given time. The product divided by 231 gives number of gallons in time named.

To find the capacity of a cylinder in gallons, multiply the area, in inches, by the length of stroke, in inches, which will give the total number of cubic inches; divide this product by 231 (which is the cubical con-

tents of a gallon in inches), and quotient is capacity in gallons. Ordinary speed to run pumps is 100 feet of piston travel per minute.

To find the quantity of water elevated in one minute, running at 100 feet of piston travel per minute, square the diameter of water cylinder and multiply by 4.

To find the diameter of a pump cylinder to move a given quantity of water or liquid per minute (100 feet of piston travel being the speed), divide the number of gallons by 4, then extract the square root, which will be the required diameter in inches.

To find the velocity in feet per minute necessary to discharge a given volume of water or liquid in a given time, multiply the number of cubic feet of water by 144, and divide the product by the area of the pipe in inches.

To find the area of a required pipe, the volume and velocity of water or liquid being given, multiply the number of cubic feet of water or liquid by 144, and divide the product by the velocity in feet per minute. The area being found, it is easy to get the diameter of pipe necessary.

The actual brake horsepower required to drive a centrifugal pump can be found by the following formula:

$$.000253 \; \frac{\text{U. S. Gallons per Minute} \times \text{Head in Feet}}{\text{Pump Efficiency Per Cent}} = \text{Brake Horsepower}$$

In case the liquid being pumped is heavier than water, it will be necessary to multiply by the specific gravity.

The horsepower of the motor selected should be approximately ten per cent greater than the brake horsepower, as centrifugal pumps are designed to be slightly over capacity.

To figure current consumption per thousand gallons of liquid or water pumped, the following formula will be found convenient:

$$\frac{\text{Head in Feet} \times .0031456}{\text{Pump Efficiency} \times \text{Motor Efficiency}} = \text{KWH's per 1000 gallons pumped.}$$

Note: When piping to engine friction losses should at all times be considered. Exhaust piping or discharge piping should be connected so that all unnecessary bends be avoided. When elbows and valves are introduced the friction is increased. Roughly the friction in a short radius 90-degree elbow can be estimated as the same as the loss in a straight pipe 30 times the diameter of the elbow in length, the friction in a long radius elbow that of a pipe 16 times as long as the diameter of the elbow and the friction of 45-degree elbows one-half that of 90-degree elbows.

To Find the Pressure in Pounds Per Square Inch of a Column of Water: Multiply the height of the column in feet by .434. (Approximately every foot elevation is called equal to one-half pound pressure per square inch.)

Doubling the Diameter of a Pipe Increases Its Capacity Four Times. Friction of liquids in pipes increases as the square of the velocity.

Ordinary Speed to Run Pumps is 100 feet of piston per minute. To find quantity of water elevated in one minute running at 100 feet of piston per minute: Square the diameter of water cylinder in inches and multiply by 4.

Example: If the capacity of a 5-inch cylinder is desired, the square of the diameter (5 inches) is 25, which, multiplied by 4, gives 100, which is gallons per minute (approximately).

Convenient Multiples:

For the circumference of a Circle, multiply the diameter by 3.1416.

For the diameter of a Circle, multiply circumference by .31831.

For the Area of a Circle, multiply square of diameter by .7854.

For the Side of an Equal Square, multiply diameter by .8862.

For the Surface of a Sphere, multiply square of diameter by 3.1416.

For the Solidity of a Sphere, multiply cube of diameter by .5236.

For the Side of an inscribed Cube, multiply the radius of sphere by 1.1547.

The Area of the Base of a Pyramid, or Cone, whether round, square or triangular, multiplied by one-third of its height, equals the Solidity.

PRESSURES CORRESPONDING TO GIVEN HEADS OF WATER

Temperature = 62° Fahrenheit

Pressure in pounds per square inch = P = .036085 h (head in inches)

h	P	h	P	h	P
0.1	.00361	0.8	.0289	6	.2165
0.2	.00722	0.9	.0325	7	.2526
0.3	.0108	1	.0361	8	.2887
0.4	.0144	2	.0722	9	.3248
0.5	.0180	3	.1083	10	.3608
0.6	.0216	4	.1443	11	.3969
0.7	.0253	5	.1804		

Pressure in pounds per square inch = P = .43302 H (head in feet)

H	P	H	P	H	P
1	.4330	25	10.83	105	45.47
2	.8660	26	11.26	110	47.63
3	1.299	27	11.69	115	49.80
4	1.732	28	12.12	120	51.96
5	2.165	29	12.56	125	54.13
6	2.598	30	12.99	130	56.29
7	3.031	31	13.42	135	58.46
8	3.464	32	13.86	140	60.62
9	3.897	33	14.29	150	64.95
10	4.330	34	14.72	160	69.28
11	4.763	35	15.16	170	73.61
12	5.196	40	17.32	180	77.94
13	5.629	45	19.49	190	82.27
14	6.062	50	21.65	200	86.60
15	6.495	55	23.82	210	90.93
16	6.928	60	25.98	220	95.26
17	7.361	65	28.15	230	99.59
18	7.794	70	30.31	240	103.9
19	8.227	75	32.48	250	108.3
20	8.660	80	34.64	260	112.6
21	9.093	85	36.81	270	116.9
22	9.526	90	38.97	280	121.2
23	9.959	95	41.14	290	125.6
24	10.39	100	43.30	300	129.9

TABLE OF GALLONS

	Cubic inches in a gallon.	Wt. of a gal. in lbs., Avoir.	Gallons in a cubic foot
United States	231	8.33	7.480
New York	221.819	8.00	7.901
Imperial	277.274	10.00	6.232

Weight of a Cubic Foot of Water, English Standard = 62.321 pounds avoirdupois.

PRESSURES OF WATER AND EQUIVALENTS (Rankine)

One Atmosphere (= 29.922 inch mercury) = 33.9 feet of water.

One inch of mercury at 32° = 1.1334 feet of water.

One cubic foot of average seawater = 1.026 cubic feet of pure water in weight.

One Fahrenheit degree = .55555 Centigrade degree.

One Centigrade degree = 1.8 Fahrenheit degrees.

Temperature of melting ice = 32° on Fahrenheit's scale.

Temperature of melting ice = 0° on Centigrade scale.

Specfiic Gravity is defined to be the ratio of the weight of a given bulk of substance, to the weight of the same bulk of pure water at a standard temperature.

HORSEPOWER TRANSMITTED BY BELTING

Width of Belt in inches	H.P. per 100 feet Belt Velocity — Single Belt	H.P. per 100 feet Belt Velocity — Double Belt	Width of Belt in inches	H.P. per 100 feet Belt Velocity — Single Belt	H.P. per 100 feet Belt Velocity — Double Belt
2	.3	----	12	2.1	3.5
3	.5	----	14	2.5	4.1
4	.7	1.1	16	2.9	4.6
5	.9	----	20	---	5.8
6	1.1	1.7	24	---	7.0
8	1.4	2.3	30	---	8.7
9	1.6	----	36	---	10.5
10	1.8	2.9	40	---	11.6

Note: To find diameter of driving pulley, multiply the diameter of the driven pulley by its revolutions and divide the product by the revolutions of the driver; the quotient will give the diameter of the driver.

HORSEPOWER REQUIRED FOR DRIVING CENTRIFUGAL PUMPS

Size of Pump	—Capacity of Pumps— Gals per Minute	Cu. Ft. per Sec. also Acre Ins. per Hr.	Miner's Ins.	Horse Power Required for —Each Foot of Lift— Theoretical H. P.	Recommended H. P.
1½	50	.1	4.5	.013	.04
2	100	.2	8.9	.025	.06
2½	150	.3	13.4	.038	.085
3	225	5	20.	.057	.114
3½	300	.6	27.	.08	.16
4	400	.9	35.7	.10	.20
5	700	1.5	62.5	.17	.34
6	900	2.0	80.	.23	.39
7	1,200	2.6	107.	.31	.50
8	1,600	3.5	143.	.41	.67
10	3,000	6.6	268.	.76	1.17
12	4,500	10.0	400.	1.13	1.75

The sizes of pumps given above correspond to the diameters of discharge openings in inches.

DISCHARGE OF WATER

In following table the amount of water to be discharged in corresponding relation to the pump unit to Diesel engine, gives a clear idea as to the amount of water a pump can discharge a stream of water.

GIVEN IN CUBIC FEET PER MINUTE, THE AREA OF THE STREAM BEING ONE SQUARE INCH IN THIS TABLE

Head	Discharge	Head	Discharge
1	3.34	39	20.87
2	4.73	40	21.13
3	5.79	41	21.38
4	6.68	42	21.64
5	7.47	43	21.90
6	8.18	44	22.15
7	8.84	45	22.40
8	9.45	46	22.65
9	10.02	47	22.89
10	10.51	48	23.14
11	11.08	49	23.38
12	11.57	50	23.61
13	12.05	51	23.85
14	12.50	52	24.08
15	12.94	53	24.31
16	13.37	54	24.54
17	13.78	55	24.76
18	14.18	56	24.99
19	14.57	57	25.21
20	14.95	58	25.43
21	15.31	59	25.65
22	15.67	60	25.87
23	16.02	61	26.08
24	16.37	62	26.29
25	16.71	63	26.49
26	17.04	64	26.72
27	17.36	65	26.92
28	17.68	66	27.13
29	17.99	67	27.33
30	18.30	68	27.54
31	18.60	69	27.74
32	18.90	70	27.94
33	19.20	71	28.14
34	19.49	72	28.34
35	19.77	73	28.53
36	20.05	74	28.73
37	20.33	75	28.93
38	20.60	76	29.11

DISCHARGE OF WATER (Continued)

Head	Discharge	Head	Discharge
77	29.30	114	35.66
78	29.49	115	35.82
79	29.68	116	35.97
80	29.87	117	36.12
81	30.06	118	36.28
82	30.24	119	36.43
83	30.42	120	36.58
84	30.61	121	36.73
85	30.79	122	36.88
86	30.97	123	37.03
87	31.15	124	37.18
88	31.33	125	37.33
89	31.50	126	37.48
90	31.68	127	37.63
91	31.86	128	37.78
92	32.04	129	37.93
93	32.20	130	38.07
94	32.38	131	38.22
95	32.55	132	38.37
96	32.72	133	38.51
97	32.89	134	38.66
98	33.06	135	38.80
99	33.23	136	38.95
100	33.40	137	39.09
101	33.57	138	39.23
102	33.73	139	39.37
103	33.90	140	39.51
104	34.06	141	39.65
105	34.22	142	39.79
106	34.39	143	39.93
107	34.55	144	40.07
108	34.71	145	40.21
109	34.87	146	40.35
110	35.03	147	40.49
111	35.19	148	40.63
112	35.35	149	40.77
113	35.50	150	40.90

THE PRESSURE OF WATER AT DIFFERENT ELEVATIONS

Feet Head	Equals Pressure per Sq. In.	Feet Head	Equals Pressure per Sq. In.	Feet Head	Equals Pressure per Sq. In.
1	.43	130	56.31	260	112.62
5	2.16	135	58.48	265	114.79
10	4.33	140	60.64	270	116.96
15	6.49	145	62.81	275	119.12
20	8.66	150	64.97	280	121.29
25	10.82	155	66.14	285	123.45
30	12.99	160	69.31	290	125.62
35	15.16	165	71.47	295	127.78
40	17.32	170	73.64	300	129.95
45	19.49	175	75.80	310	134.28
50	21.65	180	77.97	320	138.62
55	23.82	185	80.14	330	142.95
60	25.99	190	82.30	340	147.28
65	28.15	195	84.47	350	151.61
70	30.72	200	86.63	360	155.94
75	32.48	205	88.80	370	160.27
80	34.65	210	90.96	380	164.61
85	36.82	215	93.13	390	168.94
90	38.98	220	95.30	400	173.27
95	41.15	225	97.49	500	216.58
100	43.31	230	99.63	600	259.90
105	45.48	235	101.79	700	303.22
110	47.64	240	103.96	800	346.54
115	49.81	245	106.13	900	389.86
120	51.98	250	108.29	1000	433.18
125	54.15	255	110.46		

SPECIFIC GRAVITIES CORRESPONDING TO DEGREES BAUME

(Liquids Lighter than Water)

SPECIFIC GRAVITIES CORRESPONDING TO DEGREES BAUME

(Liquids Lighter than Water)—Continued

Degree Baume	Specific Gravity	Degree Baume	Specific Gravity	Degree Baume	Specific Gravity
13	0.979	40	0.824	70	0.700
11	0.993	41	0.819	71	0.697
12	0.986	42	0.814	72	0.693
13	0.079	43	0.809	73	0.690
14	0.972	44	0.805	74	0.686
15	0.966	45	0.800	75	0.683
16	0.959	46	0.796	76	0.680
17	0.952	47	0.791	77	0.676
18	0.946	48	0.787	78	0.673
19	0.940	49	0.782	79	0.670
20	0.933	50	0.778	80	0.667
21	0.927	51	0.774	81	0.664
22	0.921	52	0.769	82	0.660
23	0.915	53	0.765	83	0.657
24	0.909	54	0.761	84	0.654
25	0.903	55	0.757	85	0.651
26	0.897	56	0.753	86	0.648
27	0.892	57	0.749	87	0.645
28	0.886	58	0.745	88	0.642
29	0.881	59	0.741	89	0.639
30	0.875	60	0.737	90	0.636
31	0.870	61	0.733	91	0.634
32	0.864	62	0.729	92	0.631
33	0.859	63	0.725	93	0.628
34	0.854	64	0.722	94	0.625
35	0.849	65	0.718	95	0.622
36	0.843	66	0.714	96	0.620
37	0.838	67	0.711	97	0.617
38	0.833	68	0.707	98	0.614
39	0.828	69	0.704	99	0.611

CHAPTER XV.

BATTERIES

CARE OF A STORAGE BATTERY

In the proper care of a storage battery, there are four things to be remembered of great importance. By remembering these 75% of the battery troubles will be eliminated.

(1) Keep all cells filled with distilled water to a level ½ inch above the level of the plates. NEVER fill the cells FULL.

(2) Do not attempt to use a battery in a leaking condition.

(3) Test the gravity of all cells with a hydrometer syringe on the first and fifteenth of every month. If any cells are below 1.275 on two successive testing dates, take the battery out and have it fully charged.

(4) NEVER allow the battery to become heated above 110 degrees F. while in service. Watch the battery for heating once or twice a day during warm weather. When the top connectors become more than blood warm to the touch take the temperature with a dairy thermometer. If the temperature reaches 100 degrees burn lamps to bring down temperature. If the temperature reaches 120 degrees the battery may become ruined.

Period of Filling with Water:

A battery should be filled with pure water without fail, once every week in summer, and once every two weeks in winter.

It is very important that the solution should be at the proper height in all of the cells. This can be easily determined by looking through the vent hole through the top.

Looking into the hole exposed by removing the vent plug one sees an inner hole that looks like the bottom of the tube. If it is filled above this point, it will slop over, because when the battery is charging the solution contains thousands of minute bubbles which cause it to expand and occupy a greater space than when it is not being charged.

Effect of Overfilling:

Overfilling causes the solution to run over the top of the battery down the sides of the box. It will rot the battery case.

Water For Filling:

Water for filling a battery should be pure and free from foreign matter. This should be distilled water, melted artificial ice, or filtered

rain water which has not come in contact with iron pipes or tin roofs. Avoid the use of spring water, river and well water; they are liable to contain iron or other substances detrimental to the life of the battery.

Use Water Only:

Only water should be used regularly for filling. If acid solution is used the electrolyte will gradually get stronger resulting in eating through the plates and separators and destroying the insulation of the battery. The battery will then become dead. If the battery becomes tipped upside down, or spilled, otherwise, it may be necessary to add to the electrolyte.

Method of Filling:

In filling the battery the most convenient way is to use a hydrometer syringe. The advantage of using the hydrometer is that if you should get too much water in one cell, the excess solution can be withdrawn and used in the next cell, or to put it back into the receptacle.

How To Use Hydrometer:

In preparing to add water to the cells, first obtain pure water and fill receptacle (porcelain or glass), which is to be used for the battery. Next, take bulb and squeeze firmly in the palm of the hand to exclude the air, then place the tip of the hydrometer in the glass of water. Then, insert the tube in the vent hole of battery and squeeze the bulb until the cell is filled to the desired height, but do not over fill. Release the tube and hold it in a horizontal position so as to keep the solution from dripping on the battery and again compress the bulb and expel the balance of the contents into the glass.

What Is a Dead Battery?

A storage battery is said to be dead when it will deliver no appreciable current at a voltage sufficient to do work. A battery goes dead from one of two causes—from insufficient current, which is lack of nourishment, or from some internal trouble. In the former case all cells in the battery show low specific gravity. The remedy is charging, preferably a long continuous charge from an outside source. If the cause is internal the troublesome cell will show low gravity. In this case the battery needs attention.

Lack of Solution:

This rapidly shortens the battery's life by reducing the area of active surface on the plates. If a battery is used for any length of time with the solution below the tops of plates they will rapidly disintegrate. Regular inspection and filling with pure water is the preventative.

Under-Charging:

If too much current is taken by the lamps and other electric apparatus, or the generator is not large enough to furnish sufficient current, the battery will go dead. It must then be taken off and charged from some outside source.

Short-Circuits:

These are caused by bad insulation, wiring coming in contact with metal parts of engine and by laying a wrench, screw driver, or other tools on top of battery.

What Is Electrolyte?

The electrolyte is the solution in the battery and consists of a definite mixture of "pure" sulphuric acid and distilled or other "pure" water. The sulphuric acid must be "chemically pure" to a certain standard, which is the same standard as is usually sold in drug stores as "CP" (chemically pure), or by the chemical manufacturers as "battery acid." Do not confuse "chemically pure" sulphuric acid with sulphuric acid of "full strength," because the use of the latter would not materially reduce the strength, but would make the mixture impure and endanger the efficiency of the battery.

Full strength of concentrated sulphuric acid is a heavy, oily liquid having a strength (specific gravity) of about 1.835. If put into the battery, it would quickly ruin it, and must, therefore, first be diluted with "pure" water to the proper strength for the particular type of battery to which it is to be added.

If electrolyte of the proper strength is not on hand, it may be prepared from "chemically pure" sulphuric acid by mixing the acid with pure water. The acid may be of any strength, providing it is stronger than the electrolyte desired. The proportions of acid and water depend upon the strength of the acid. When mixing, take the following precautions:

1. Use a glass, china, earthenware, rubber or lead vessel; never metallic, other than lead.

2. Carefully pour the acid into water; not the water into the acid.

3. Stir thoroughly with a wooden paddle and allow to cool before taking a hydrometer reading of the strength.

Electrolyte, like most substances, expands when hot, affecting the hydrometer reading. To compare different hydrometer readings, therefore, the temperature should be the same. It is not necessary, however, to actually bring the temperatures to the same value, because it is a known fact that every three degrees increase in temperature decreases the hydrometer reading one point, and this fact can be used in estimating what the hydrometer reading would be at a normal temperature. The normal is taken as 70 degrees Fahrenheit. If the hydrometer reading at 100 degrees is 1.270, it would be 10 points more, or 1.280 at 70 degrees. If the reading is 1.290 at 40 degrees, it would also be 1.280 at

70 degrees. Therefore, although the two actual readings differ by 20 points, the difference is all due to temperature, and if the temperature were the same, the readings would be the same. When the temperature is much above or below normal, the hydrometer readings should be "corrected for temperature."

Why Is the Electrolyte Weaker in Batteries in Tropical Climates?

The electrolyte used in such batteries is purposely made weaker because batteries operated in tropical climates give better results if the solution is weaker than that used for batteries in colder climates. Places where freezing of water never occurs are regarded as having tropical climates.

How Can Trouble Be Located?

1. Go over all connections. A loose or dirty connection is often the cause of trouble. If the connections between the battery and cable terminals are not kept well coated with gasoline, they may corrode, causing a poor connection, or else opening the circuit all together. If the connector is causing the trouble, remove it and clean the parts thoroughly with weak ammonia. Then remove all dirt, apply vaseline, tighten the connection perfectly and give the whole connection a heavy coating of vaseline.

2. There may be a leak or ground in the wiring. Test for this by turning all lamp switches and then removing the bulbs from the sockets. Disconnect one of the cables at the battery and in its place tightly hold a file against the battery post, making sure there is good electrical contact between the file and the post. Then rub the cable terminal along the file; if sparks are noticed, there is a ground in the wiring, which must be looked for and removed.

3. If the generator of the starting system is not in proper adjustment, the battery will not be kept supplied with the proper amount of current. If the supply is insufficient, the battery will become discharged; if it is too much, the battery solution will become hot (110 degrees Fahrenheit). The generator should be readjusted to deliver more or less current, as the case requires.

When Is the Best Time to Add Water?

In warm weather, it makes no difference when water is added. In freezing weather, it should be added just before using the engine. The reason is that water will remain on top of the solution until it is mixed with it by action of the battery. If not mixed with the solution, it would freeze almost as quickly as outside the battery.

Why Do Hydrometer Readings Indicate the Condition of a Battery?

When current is taken from a battery, a certain part of the solution combines with the plates, leaving the solution weaker. When current is

put back into the battery, this is returned to the solution which is strengthened again. A measurement of the strength of the solution, therefore, will indicate the condition of the battery, because when the battery is fully charged the solution will be strong and when it is discharged the solution will be weak.

Can a Battery Freeze?

The freezing point of the battery solution depends upon its strength. For example, a solution with a strength or specific gravity of 1.250 will not freeze until the extremely cold temperature of 62 degrees Fahrenheit below Zero is reached. A strength of 1.150 will freeze at 5 degrees above Zero, so it will be seen there is little danger of freezing except with a completely discharged battery. Moreover, at these freezing points, the solution is slushy and does not become hard until the temperature goes still lower. If water is added to a battery in freezing weather and then not stirred in with the solution by charging the battery, it will remain on top of the solution and may freeze. But to avoid this possibility, warning is given not to add water in cold weather until just before running the engine.

DEFINITION AND DESCRIPTION OF TERMS AND PARTS OF A STORAGE BATTERY

Acid: The active component of the Electrolyte.

Active Material: The active portion of the battery plates; the most used is peroxide of lead on the positive and spongy metallic lead on the negative.

Alternating Current: Electric current which does not flow in one direction only, like direct current, but rapidly reverses its direction or "Alternates" in polarity so that it will not charge the battery.

Ampere: The unit of measure of the rate of flow of electric current.

Ampere Hour: The unit of measure of the quantity of electric current. Thus, 2 amperes flowing for one-half hour = 1 Amp, hr.

Battery: A number of complete cells assembled in one case.

Buckling: Warping or bending of the battery plates.

Burning Strip: A convenient form of lead, in strips for filling up the joints in making burned connections.

Case: The containing box which holds the battery cells.

Cell: The battery unit, consisting of an element complete with electrolyte, in its jar with cover.

Cell Connector: The metal link which connects the positive post one cell to the negative post of the adjoining cell.

Charge: Passing direct current through a battery in the direction opposite to that of discharge, in order to put back the energy used on discharge.

Charge Rate: The proper rate of current to use in charging a battery from an outside source. It is expressed in amperes and varies for different sized cells.

Corrosion: The attack of metal parts by acid from the electrolyte; it is the result of lack of cleanliness.

Cover: The rubber cover which closes each individual cell; it is flanged for sealing compound to insure an effective seal.

Discharge: The flow of electric current from a battery through a circuit. The opposite to "charge."

Electrolyte: The fluid in a battery cell, consisting of specially pure sulphuric acid or rather chemicals diluted with pure water.

Element: One opposite group and negative group with separators, assembled together.

Filling Plug: The plug which fits in and closes the orifice of the filling tube in the cell cover.

Flooding: Over-flowing through the filling tube.

Gasing: The bubbling of the electrolyte caused by the rising of gas set free toward the end of the charge.

Generator System: An equipment including a generator for automatically recharging the battery, in contradict action to a straight storage system, where the battery has to be removed to be recharged.

Gravity: A contraction of the term "Specific Gravity," which means the density compared with water as a standard.

Grid: A set of plates, either positive or negative, joined to a strap. Groups do not include separators.

Hold Down Clips: Brackets for the attachments of bolts for holding the battery securely in position.

Hydrogen Flame: A very hot and clean flame of hydrogen gas and compressed air, used for making burned connections.

Hydrogen Generator: An apparatus for generating hydrogen gas for lead burning.

Hydrometer: An instrument for measuring the specific gravity of the electrolyte.

Hydrometer Syringe: A glass barrel enclosing a hydrometer and provided with a rubber bulb for drawing up electrolyte.

Jar: The hard rubber container holding the element and electrolyte.

Lead Burning: Making a joint by melting together the metal of the parts to be joined.

Lug: The extension from the top frame head plate, connecting strap to strap.

Maximum Gravity: The highest gravity which the electrolyte will reach by continued charging indicating that no acid remains in the plates.

Oil of Vitriol: Commercial name for concentrated sulphuric acid (1.835 specific gravity). This will ruin a battery if used.

Plates: Metallic grids supporting active materials. They are alternately positive (brown) and negative, (gray).

Polarity: Electrical condition. The positive terminal of a cell or battery, or the positive wire of a circuit, is said to have positive polarity; the negative, negative polarity.

Post: The portion of the strap extending through the cell cover, by means of which connection is made to the adjoining cell or to the position for light or ignition.

Rectifier: Apparatus for converting alternating current into direct current.

Resistance: Material (usually lamps or wire) of low conductivity inserted in a circuit to retard the flow of current. By varying the resistance, the amount of current can be regulated.

Rubber Sheets: Thin, perforated hard rubber sheets used in combination with the wood separators in some types of batteries. They are placed between the grooved sides of the wood separators and the positive plate.

Sealing Compound: The acid proof compound used to seal the cover of the jar.

Sediment: Active material which gradually falls from the plates and accumulates in the space below the plates provided for that purpose.

Separators: Sheets of grooved wood, specially treated, inserted between the positive and negative plates to keep them from contact.

Short Circuit: A metallic connection between the positive and negative plates within the cell. The plates may be in actual contact or material may be lodged and bridged across. If the separators are in good condition, a short circuit is unlikely to occur.

Spacers: Wood strips used in some types to separate the cells in the case, and divided to provide a space for the tie bolts.

Specific Gravity: The density of the electrolyte compared to water as a standard. It indicates the strength and is measured by the hydrometer.

Starvation: The result of giving insufficient charge in relation to the amount of discharge, resulting in poor service and injury to the battery.

Strap: The leading casting to which the plates of a group are joined.

Sulphated: The condition of plates having an abnormal amount of lead sulphate caused by "starvation" or by allowing the battery to remain discharged.

Terminal Connectors: Devices attached to the positive posts to one end of the cell and the negative of the other, by means of which the battery is connected to the ignition and lighting circuit.

Tie Bolts: Bolts, which in some types, extend through the battery case between the cells and clamp the jars in position.

Top Nuts: The hexagon nut, which, in batteries with bolted connections, screws on the post and holds the connectors and sealing nuts in place.

Voltage: Electrical potential or pressure, of which the volt is the unit.

CONDUCTORS AND INSULATORS

Good Conductors

Silver	Tin
Copper	Lead
Aluminum	German Silver (copper 2 parts, zinc 1, nickel 1)
Zinc	Platinoid (German silver 49 parts, tungsten 1 part)
Brass (according to the composition)	Antimony
Platinum	Mercury
Iron	Bismuth
Nickel	

Fair Conductors

Charcoal and Coke	Acid Solutions
Carbon	Living Vegetable Substances
Plumbago	Moist Earth

Partial Conductors

Water	Pine
The Body	Rosewood
Flame	Lignum Vitae
Linen	Teak
Cotton	Marble
Mahogany	

Non-Conductors or Insulators

Slate	Gutta Percha
Oils	Shellac
Porcelain	Ebonite
Dry Leather	Amber
Dry Paper	Paraffine Wax
Wool	Glass (varies with quality)
Silk	Mica
Sealing Wax	Jet
Sulphur	Dry Air
Resin	

QUESTIONS AND ANSWERS FOR MAINTAINING BATTERIES

Q. What is a volt?

A. A volt is a unit of pressure. Current will not flow without there is a difference of potential or pressure, and this difference is measured in volts. It is analogous to head in pipe in hydraulics.

Q. What is an ampere?

A. An ampere is a unit of current and signifies the rate of flow of electricity. The analogous term in hydraulics would be gallons per minute.

Q. What is an Ohm?

A. An Ohm is the unit of resistance or friction. It is analogous to friction head in pipe in hydraulics.

Q. What is a Watt?

A. A Watt is the unit of power and it is volts multiplied by amperes.

Q. What is a kilowatt?

A. A Kilowatt is 1000 watts.

Q. What is meant by a "Kilowatt" hour and a "Watt" hour?

A. A Kilowatt hour and a Watt hour is the product of the product of the number of watts or kilowatts respectively by the number of hours.

Q. What is a dynamo?

A. A dynamo is a machine for converting electrical energy into mechanical energy.

Q. What is a direct current dynamo?

A. A direct current dynamo is a machine for generating a current flowing only in one direction. (This current used for charging batteries.)

Q. What is an alternating current dynamo?

A. An alternating current dynamo is one generating a current which rapidly reverses its direction of flow. (This current is generally used for lighting service and domestic use.)

Q. What is a primary battery?

A. A primary battery is a combination of metals and acids which generate an electric current by chemical action.

Q. What is a secondary or storage battery?

A. A storage battery is one which, after current has passed through, is capable of giving off current when its terminals are connected.

Q. What is the distinction between a battery and a battery cell?

A. A battery of any kind is made up of a number of divisions. In an electric battery these divisions are called cells and consist of an element containing one negative and one positive terminal.

Q. What governs the number of cells in a battery?

A. The required volts and amperes as compared with the volts and amperes per cell.

Q. How are the terminals of a battery designated?

A. Each battery or cell has a positive or negative terminal, the current flowing externally from the positive to the negative.

Q. How are the battery cells connected to combine voltage?

A. In series, that is, the positive to the negative of the next and the positive and negative terminals at either end of the series to the load.

Q. How are battery cells connected to combine amperage?

A. In parallel, all positives together and all negatives together, and load taken off from the positive and negative connection.

Q. How are battery cells connected to combine both volts and amperage?

A. By connecting a number of cells in series to get the required voltage and a sufficient number of these combinations in parallel to give the proper amperage.

Q. How are incandescent lamps rated?

A. In candlepower or watts.

Q. How do you determine the amperes required to operate a lamp?

A. Select the w.p.c. according to class. Multiply by candlepower and divide by voltage of current.

Q. How do you determine the number of lamps a dynamo will operate?

A. Multiply kilowatt rating by 1000 watts. Divide by wattage of lamps.

Q. How do you determine the number of lamps a battery will operate for a given number of hours?

A. Divide the ampere hours battery capacity by the sum of the product of the amperes of each lamp by the hours it will operate.

Q. How can a size of dynamo be determined to charge a battery?

A. Multiply the ampere charging rate by the voltage of the battery, which gives generator capacity in watts.

Q. Can a storage battery be charged with alternating current?

A. No. It requires direct current or the alternating current must be changed by a rotary conveyor or a rectifier.

Q. A battery giving 120 volts was to be charged, and you had a generator of the same voltage, could you charge from the same?

A. This can be done by dividing the battery into two sections of 60 volts each and connecting the sections in series. In such a case the current capacity of the generator must be double the charging capacity of the battery.

Q. What would the object be of such a plant?

A. The lights can be operated from either, the battery or the generator direct.

DEFINITION OF SPECIFIC GRAVITY

Water is universally adopted as the standard by which the relative weight of all liquids and solids are determined, this relation being expressed by the term "specific gravity." The specific gravity of a body, therefore, indicates its weight as compared with that of an equal body in the form of volume of pure water. Determinations of specific gravity generally referred to the weight of one cubic foot of water at sixty-two degrees Fahrenheit. At the more important temperatures the weights are as follows:

Weight of One Cubic Foot of Pure Water:

At 32	degrees F. (freezing point)	62.418 lbs.
At 39.1	degrees F. (maximum density)	62.425 lbs.
At 62	degrees F. (standard temperature)	62.355 lbs.
At 212	degrees F. (boiling point (under atmos. pressure)	59.640 lbs.

For general purposes the weight of water is taken in round numbers as 62.5 pounds per cubic foot. Bulk water is usually measured by the gallon. The volume of which is 231 cubic inches (the British gallon contains 277.274 cubic inches), or 0.134 cubic feet. A gallon of water at sixty-two degrees, therefore, weighs 8.35 and 7.48 gallons equals one cubic foot.

Pressure of Water:

From the weight of water at the standard temperature of 62 degrees, its pressure upon any exposed surface may be readily determined for any given depth or head. The weight of one cubic foot at the above temperature being 62.355 lbs., it is evident that for a head of one cubic foot the pressure must be 62.355 lbs., per square foot, and $\frac{62.355}{144} =$ 0.433 lbs. per square inch; and, further, that a pressure of one pound per square inch will be produced by a head of

$$\frac{1}{0.433} = 2.309 \text{ feet.}$$

CARE AND MAINTAINING OF THE LEAD ACID BATTERY

In writing this treatise on storage batteries it will be undertaken to explain and make as simple as possible the theory, care, and maintenance of the lead acid battery.

First, let us understand for once and all that a storage battery does not mean that electricity is stored as the name storage battery would seem to imply.

All minerals, gases, and liquids are classed either as a positive (+), or negative (—) element.

In the lead acid cell we have the sponge lead plate which is a highly positive element and the lead peroxide plate which is one of the most negative elements known.

Next, the electrolyte which is any chemical fluid or semi-fluid, which will be decomposed by the passage of an electrical current.

Now it follows that if these two elements, the P B sponge lead and the Pb02 = lead peroxide, one electric positive, the other electro negative, be put in an electrolyte composed of dilute sulphuric acid, we have the two elements of different polarity in an electrolyte and as electricity closely resembles water in a pipe always tending to flow from a higher to a lower level, so electricity always tends to flow from + to —, but, like water, if it has no outlet it will cease to flow.

As the current flows from the Pb plate to the PbO^2 plate it leaves the Pb02 plate at the terminal of this plate and continues through the conductor, which is furnished for it, and back to the Pb plate, this, as can readily be seen, makes the terminal of the Pb02 plate a positive terminal, since the current leaves the cell by this terminal; so it also makes the terminal of the Pb plate a negative terminal of the Pb plate a negative terminal since it enters the cell again by this terminal. The positive terminal is on what voltically is termed the negative plate, and the negative terminal is on what is voltically termed the positive plate. Since great confusion arises in the mind of some students as to the relation of the plates, the plate by which the current leaves the cell on discharge is always known as the positive. The positive plate of a lead acid cell is of a dark brown almost chocolate color and is very firm when the plate is in good condition, while the negative is gray and is soft and springy, if in good condition.

As the current flows from the Pb plate through the H^2SO^4 = sulphuric acid and water to the PbO^2 plate due to the potential difference of these elements it decomposes the H^2SO^4. The plates taking the SO^4 from the electrolyte and combining it with the lead, and, since lead and sulphuric acid combined gives lead sulphate, both plates taking in combination with them the SO^4 from the electrolyte, changes them both to the same material theoretically, so, of course, no more potential difference exists between them, and there will be no more current flow, in the condition the battery is discharged. As sulphuric acid tends to harden and deteriorate any mineral with which it comes in contact with one should have only enough of such material in the electrolyte to combine with the lead in both slates to turn them to lead sulphate upon discharge.

So in a cell in which there is plenty of room for electrolyte a lower gravity can be made than in one where there is but little space for electrolyte, since we can have plenty of SO^4 to combine with the plates and still not have it so concentrated as to do damage to the elements of the battery. On the other hand, in a cell where the space for the electrolyte is very limited we must have a higher specific gravity in order to have

the necessary SO^4 to combine with all of the active material of the plates to complete their discharge.

So from the preceding one can readily see that the capacity of a storage cell depends upon the amount of active material in the plates that can combine with SO^4 of the electrolyte and the amount of SO^4 of the electrolyte that can combine with the plates to change them to $PbSO_4$,—lead sulphate.

If the symbols used in the writing have been noted a great step toward an understanding of the lead acid storage cell has been made, but so as to make it plainer we shall put down the symbols and their meaning.

Pb = Sponge lead, negative plate.

PbO^2 = Lead peroxide or positive plate.

H^2SO^4 = Water H^2O. Sulphuric acid SO^3 the two combined forming the electrolyte.

The chemical changes in the cell are as follows:

All fully discharged bath plates are $PbSO_4$ or lead sulphate due to the electricity decomposing the SO^4 from the electrolyte and combining it with the lead of the plates, leaving the electrolyte H^2O.

On charge the current being in the opposite direction decomposes the SO_4 from the plates back into the electrolyte changing it back to H^2SO^4 and leaving the plates in their original condition as a voltic couple.

It stands to reason since the passage of a current of electricity decomposes the H^2SO^4 combining the SO^4 with the plates on discharge, that the heavier the current the more rapid will be the change of the plates to lead sulphate or $PbSO_4$, and as the plates are porous the capacity of the battery depends upon the diffusion of the SO^4 to the innermost particles of the active material. Now sulphate is of high resistance, so, when a cell is discharged at a very high rate the sulphation will form on the surface of the plates and clog the pores of them so it will be hard for the SO_4 to penetrate to the active material in the inside of the plate, consequently, the voltage of the cell drops to its safe limit before the inside of the plates are discharged. But upon the discontinuance of the discharge the voltage will rise due to the slow diffusion of the SO^4 to the unworked portion of the active material.

Due to the above fact the capacity of a storage cell is very low at high rates of discharge.

On the other hand if the discharge be carried on at a low rate the capacity will be much greater due to the fact, that the sulphation is going on in the innermost recesses of the active material because the SO^4 has a chance to reach and combine with the inside active material the same as the outside particles of active material.

For the two above reasons a standard for time and rate in amperes has been established, the time being the greatest number of amperes that can be taken from a battery for a continuous 8 hours.

For instance, if we say a battery has an ampere hour capacity of 200 we mean that 200 ampere hours can be obtained from the battery in 8 hours time, and, since 1 ampere flowing for one hour = 1 ampere hour, 200 ampere hours would mean 200 amperes flowing for 200 hours, but since neither of the above are carried to extremes the time set to compute ampere hour capacity has been set at 8 hours.

The 200 amperes divided by the 8 hours gives us the current that will be maintained for the 8 hours, or 25 amperes.

For safe working limits a discharge must be stopped before the voltage drops so low, as to show that sulphation has been too dense in the plate, so there is a voltage limit to allow for each cell; at the 3 hour rate let the limit be 1.5 volts due to the fact, that it will rise again as soon as the discharge has stopped because there is still SO^4 in the electrolyte and Pb and PbO^2 in the inside of the plates. While at the 8 hour rate the discharge should be stopped when the voltage reaches 1.7 volts per cell, due to the fact, that the plate is sulphated all the way through and will not recuperate much.

When we speak of the ampere hour efficiency of a battery we mean the number of ampere hours that can be gotten from a battery, divided by the ampere hours which must be used to decompose the SO^4 from the plates back into the H^2O, bringing the electrolyte back to H^2SO_4, and leaving the plates Pb and PbO^2 the same as before discharged.

On account of the resistance offered by the SO^4 in the plates there is some of the charging current lost in heat and gasing and a battery with 85 per cent efficiency is considered very satisfactory, this allows for 15 per cent loss in heating and gasing in overcoming resistance due to sulphate and counter voltage.

As the battery nears a charged condition the sulphation decreases and the counter E. M. F. or counter voltage increases. This is readily seen, due to the fact, that sending a current, through the battery in a direction opposite to discharge bucking a voltage, is being bucked which tends to cause a current to flow in direction of discharge.

Therefore, the charging voltage must be higher than the voltage of this battery and as the battery nears a charged condition the current will taper due to the resistance offered up by the counter E. M. F., unless the charging voltage, as the charge nears completion. It is a very good plan to allow the current to fall very low toward the end of charge as any current that is not used in breaking down sulphation is really wasted, so as the rate of charge falls off, do not raise charging voltage unless it falls to zero or the ammeter starts showing on discharge side. In storage battery work it is always convenient to use a two-way ammeter so as to show charge and discharge without having to change connections on the meter.

Electrolyte of from 1.240 to 1.280 specific gravity, depending upon the space available in the cell for electrolyte, should be used.

As the SO^4 is the heavier elements in the electrolyte it will be noted as the cell is discharged the weight as sp. gr. of the electrolyte becomes less, due to the fact that the S04 is combining with the plates, also it will be found that the plates are becoming heavier, due to the SO^4 combining with them.

It is safe in allowing about 80 points drop in sp. gr. where 1.280 is used from these two, proper judgment must be used. No set rule as to the range of gravity need be followed as one is safe if the voltage limit is regarded. Always take voltage reading while current is flowing as it will give a better idea when to stop discharge and avoid damage caused by excessive sulphation to plates.

In mixing sulphuric acid and water to make electrolyte, always remember to pour the acid into the water, never pour the water into the acid. Always be sure the water is chemically pure. Always be sure the acid has been made for electrolyte. Always use a container that will not be attacked by acid such as hard rubber, glass or pure lead, and be sure the mixing tank is clean.

Concentrated sulphuric acid has a sp. gr. of 1.835, while distilled water is 1.000, and a very good formula for figuring out how much acid and water to use for different gravities is as follows:

Say you wanted to use 1.260 sp. gr.

Subtract 1.000 from 1.835, which gives you 835 parts of water; next, subtract 1.000 from 1.260, which will give you 260 parts of acid. This will not give you the exact specific gravity required, but is close enough so that when you have mixed nearly enough electrolyte let it cool and then after thoroughly stirring with a clean stick add enough water and acid to bring the specific gravity to the right density which can be done without much heating of the electrolyte.

Suppose you receive your battery without electrolyte which is often the case where the shipment is of long distance. The first thing to do is to determine how to assemble it without injury to the parts.

First, we will consider the separators which are placed between the plates of opposite polarity. They may be of perforated rubber so as to allow the free circulation of the electrolyte, or wood which is very porous and especially treated to withstand the effect of the acid. If of wood they will be packed in a water-tight sealed receptacle and very damp, and if they are not to be used immediately upon arrival they should be either left packed, and, if opened, placed in distilled water so as not to dry out and crack as their use is to keep the plates of opposite poles from touching one another, and if cracked, will allow the massing across of the negative plate, short circuiting the cell, as all the positive plates are connected to a common brass bar as are also all of the negatives. Thus you can readily see the effect would be the same as connecting the terminals together, thus causing a current to flow from the Pb plate to the PbO^2 plate through the electrolyte and from the PbO^2 back to the Pb through the short circuit where the plates were touching each other.

Next, look for data on the name plate of the battery to determine what the ampere hours capacity of the battery is, and what specific gravity electrolyte is to be used. After mixing electrolyte to proper gravity set aside to cool to a temperature of 80 degrees F. while you get the battery ready to accept the electrolyte.

Next, if the plates are not to be burned to their fuses interleave them, laying down a negative first, then a separator, then a positive, then another separator, then another negative, until the necessary number of plates for the cell are in position, there will be always one or more negative than positive, consequently the element will have a negative for each outside plate.

Next, place the element in a vice, being sure to place a piece of hard wood on each side next to the jaws of the vice so as to brace the plates and, also, to protect the plates from coming in contact with the iron, thus getting impurities into the plates.

Now place asbestos strips between the hanging bars so as to insure not getting any lead down into the plates, and also to insure the brass bars are high enough so as not to touch the plates of opposite polarity, also all of the positives having their hanging bars burned to a bus at one end and all of the negatives having their hanging bars burned to a bus at the opposite end of the element. After this is done with clean strips of pure lead, being sure to fuse each hanging bar to its bus bar, take a piece of brass pipe with an inside diameter of the size of the terminal you wish to use, and using it as a mould, fuse lead in it to the bus, being sure each drop of lead fuses before putting any more into the pipe, and be careful to get the terminal in a position on the busses so that when you put the element in the jar the holes in the cell cover will come right for the terminals.

Now put the elements in the jar, put the cover on, noting and marking, which is positive and which is negative. After carrying out the above instructions for each cell place them in a position so that the terminal of one cell will have next to it and burned to that of the next cell, following out this method will give you a positive terminal free for the line at one end and a negative terminal free for the line at the other end.

This is known as a series grouping of cells and has the advantage of voltage of one cell and the number of cells in the group, but it has the disadvantage of only having the ampere output of one cell because the same current that is causing a decomposition in all, consequently they will all have their plates changed to $PbSO^4$ with the ampere output of one cell.

This grouping of cells is used where it is necessary to overcome a high resistance when the load is low.

Next, put hot battery seating compound around the edge of the cover of the cell and around the terminals where they come through the cover of the cell.

Now we are ready for the electrolyte, which, when cooled down to the temperature of the surrounding atmosphere at 80 degrees F., pour it into the cell until it is well over the top of the plates, let the battery set for about twelve or fourteen hours by which time you will have found the specific gravity has fallen almost to the gravity of the water. This is caused by the SO^4 having an affinity for the dry plates.

Now start what is known as the initial charge. There is no set rule for the rate of charging current at which this may be carried on but watch the temperature rather than the current and if it rises above 110 degrees F., the rate, until it holds its temperature. It will require from three to five times the rated ampere hours capacity of the cell to complete this first charge. If the electrolyte gets down to the tops of the slats do not add electrolyte, but distilled water. When the charge is completed the specific gravity will be nearly, if not the same as when it was put in cells, and each cell should have a voltage of about 2.5 when charge is finished. Now we will try to explain, as simply as possible, the different grouping of cells and the advantage of one over the other.

First, let us get a series grouping and its advantages. We will not consider the internal resistance of the cell as it is constantly changing and would be very hard to keep track of, besides this is more lengthy than first intended it should be and we want to make you understand, if we can, without going any deeper and confusing you any more than possible.

Each lamp, motor or electrical appliance is marked for the required voltage and if you understand the different grouping of batteries you can readily see what connections to use to the best advantage.

We have 6-12 volt lamps and only 6 cells of 12 ampere hours capacity each, each cell has an average voltage of 2 volts. Now since a series grouping of cells gives us the voltage of one cell and the number in series, and each cell has an E. M. F. of two volts. We will have to connect the six cells in series in order to get the required E. M. F. Now suppose each lamp used ½ ampere, the six would use 6×5 or 3 amperes, now if they were to burn for one hour steady it would mean that 3 ampere hours had been discharged from the battery and if they were left lighted for 4 hours you would notice that they burned with less brilliancy than at first. This would be due to the plates of the battery combining with the SO^4 the electrolyte thus losing their difference of potential and consequently their electrical pressure in forcing the necessary ½ ampere through each lamp.

Since the same three amperes have been flowing through all the cells it stands that they shall all discharge with the same amount of sulphation in their plates.

Now suppose the lamps in question were 2 volt lamps, it would only be necessary for us to get the E. M. F. of one cell, but by the

proper connection we could secure the ampere hour capacity of all the cells or the capacity of one (times) the number in the group in this case, since 1 cell has 12 ampere hours the 6 would have 6×12 or 72 ampere hours capacity. We would have in the case of the 3 amperes from each cell, the SO^4 from the electrolyte combining with the plates 1/6 as fast, or the group would last 6 times as long as the series grouping before discharged or the plates sulphated.

Now suppose the lamps in question were 6 volts and we wanted the greatest capacity possible from the 6 cells.

It would be necessary to connect 3 cells in series to get the proper E. M. F. and by making two groupings of 3 cells each and connecting the 2 groups in multiple we get the necessary E. M. F. and the capacity of the 2 cells or $2 \times 12 = 24$ ampere hours, since one half of the discharge rate goes thru each series group of 3 cells, you can see the result of this multiple series grouping of cells over the series or multiple grouping by theirselves.

In the next grouping we will use 4 volt lamps.

In this grouping it will be necessary to use 2 cells only to get the required E. M. F. so by making 3 groups of 2 cells each it can be seen that only 1/3 of the discharge current will be causing the plates of each group to sulphate, consequently we get the ampere hours capacity of 3 cells or 36 ampere hours. This grouping is called series multiple.

THE EDISON STORAGE BATTERY

A storage battery is commonly looked upon as a receptacle in which to store electricity. Electricity is not a concrete matter. In fact, nobody knows just what it is, therefore, in the general apprehension of the term, it is not stored. Electricity simply causes a chemical change to be effected in certain substances, when it is caused to flow through them. These substances, in endeavoring to return to their original state, produce electricity.

The fundamental principle of the Edison Storage Battery is the oxidation and reduction of metals in an electrolyte which neither combines with nor dissolves either the metals or their oxides. Also an electrolyte which, notwithstanding its decomposition by the action of the battery, is immediately reformed in equal quantity, and is, therefore, a practically constant element without change of density or conductivity over long periods of time.

The chemical reactions in charging the Edison Storage Battery are, (1) the oxidation from a lower to a higher oxide of nickel in the positive plate, and (2) the reduction from ferrous oxide to metallic iron in the negative plate. The oxidation and reduction are performed by the oxygen and hydrogen set free at the respective poles by the electro-

lytic decomposition of water during the charge. The charging of the positive plate is, therefore, simply a process of increasing the proportion of oxygen to nickel. The proportions of nickel to oxygen in definite oxides of nickel are as follows:

	Atomic	Proportions	By	Weight
	Ni	O	Ni	O
Ni	1	1	1	.273
Ni^3O^4	1	1.33	1	.364
Ni^2O^3	1	1.5	1	.409
NiO_2	1	2	1	.545

The relative amount of oxygen necessary to oxidize nickelous oxide, or NiO, which is the oxide corresponding to the green nickel hydrate used in making the battery, to the various oxides, are given in the three reactions:

$$(1)\quad 6\ NiO + 2O = 2\ Ni^3O^4$$

$$(2)\quad 6\ NiO + 3O = 3\ Ni^2O^3$$

$$(3)\quad 6\ NiO + 6O = 6\ NiO^2$$

The NiO^2 is capable of reacting with NiO according to the reaction $NiO^2 + NiO = Ni^2O^3$.

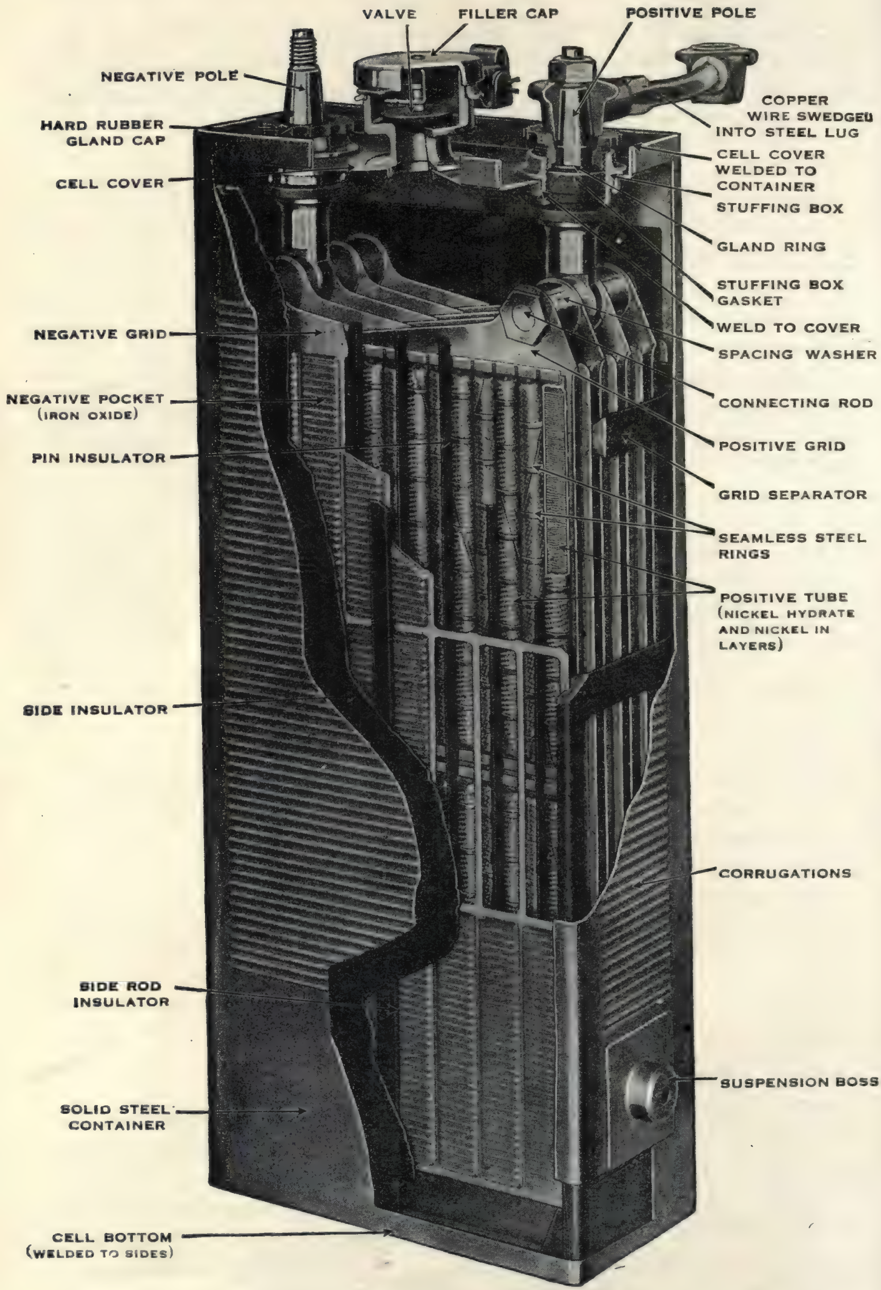
Note: Ni^3O^4 is considered as a combination of $NiO + Ni^2O^3 = Ni^3O^4$.

From a chemical standpoint a charged condition of the cell would, therefore, be represented in the positive plate by an atomic ratio of nickel to oxygen of at least 1:1.5 (or Ni^2O^3), depending on the charge. A discharged condition would be represented by a ratio of 1:1.33 (Ni^3O^4) or lower, depending on the discharge.

The discharge of the cell is simply the reversal of the above reactions, the hydrogen reducing the higher oxides of nickel to lower oxides and the oxygn oxidizing the iron to ferrous oxide.

GENERAL DESCRIPTION

For a Type A4 Edison Cell four positive plates are mounted on a steel rod, to which has been attached a vertical pole, the plates being equidistantly spaced by means of steel washers. Similarly are mounted five negative plates. "Intermesh" the four positive and five negative plates so that they will be alternately negative and positive, keep these plates from touching by putting hard rubber rods or pin insulators between them, fit hard rubber ladder pieces or grid separators to the edges of the plates, and the elements are assembled and ready to place in their container.

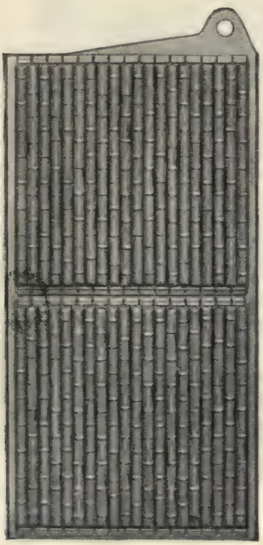


Edison Alkaline Storage Battery. This is the Only Storage Battery That Has Iron and Steel In Its Construction and Elements

The container is made of nickel-plated sheet steel with sides corrugated to increase its strength. The single side seam is welded by the oxy-acetylene blowpipe and the tops and bottom welded on in the same manner.

The assembled elements are now placed in the container with thin sheets of hard rubber on the sides not already insulated by the grid separators. A hard rubber washer is dropped on each of the vertical pole pieces, and the cell is ready to have the top welded on. It is not necessary to see the inside of the cell again.

The fittings through which the vertical poles pass are provided with soft rubber washers, and with rings and gland caps for expanding these soft rubber washers to form a gas-tight and liquid-tight packing between the top of the container and the poles.



Positive Plate

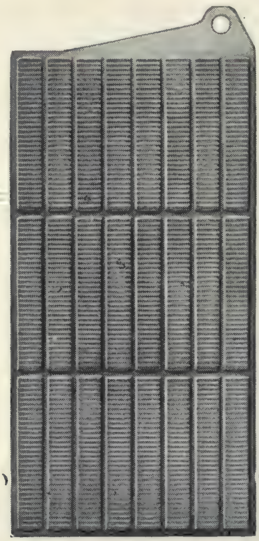
The top of a Type A4 Cell with cover welded on is shown in the illustration. The aperture for putting in solution, or adding distilled water is in the valve box in the center. In the top of the lid is a little valve which allows the gas generated during charging to get out, but no impurities or air can get in.

When any storage battery is charged, hydrogen gas forms on the negative plates and oxygen gas on the positive. These gases, in the form of minute bubbles, rise to the surface of the solution and, being lighter than air, float away. Being formed in and, subsequently, passing through the solution these minute bubbles each convey a small particle of whatever chemical the solution is composed; if they are formed in

a lead-sulphuric acid battery, sulphuric acid is the cargo; if in an Edison Alkaline Battery, potash.

When these bubbles rise from the surface of the electrolyte and come in contact with an object, they either remain until evaporation disintegrates them and deposit their cargo of acid or alkali, or they burst and accomplish the same result.

The vent of the Edison Cell is the check valve described above. To get out, the gases must lift this valve, by pressure formed within the otherwise hermetically sealed steel containing can. In doing so, a great majority of the little bubbles are burst, and the potash drains back into the cell. A few of them get by and float harmlessly away.



Negative Plate

The active material of the positive plate is nickel hydrate, which is packed in thin layers under heavy pressure in perforated steel tubes. Between the layers of nickel hydrate are still thinner layers of pure nickel. These metallic layers are made up of small flakes of nickel, each flake being about 1/16 inch square and much thinner than tissue paper.

During charge and discharge of the battery the passage of the electric current alternately oxidizes and reduces the nickel hydrate. The metallic nickel acts as a conductor, forming a path of low resistance to all parts of the active layers of hydrate.

The negative plate consists of a steel grid which supports a number of flat, perforated steel containers or pockets. The active material is iron oxide, which is held within the pockets. The pockets, after being placed in the openings of the grid, are subjected to hydraulic pressure

of one hundred and twenty tons, which forces them into permanent contact with the grid.

The passage of current during operation of the battery causes the active material of the negative plate to be alternately reduced and oxidized according as the battery is charging or discharging.

The electrolyte is a solution of caustic alkali. Unlike that of other batteries the Edison battery solution does not vary appreciably in specific gravity during the cycle of charge and discharge. Another distinctive and favorable feature of the Edison battery is that the electrolyte is a "preserver" of steel and nickel and renders impossible many of the diseases encountered in storage battery practice.

The finished cells are mounted in wooden trays and are connected to another by copper connectors provided with tapered steel lugs. Each cell is supported by rubber buttons imbedded in the tray slats. The cells are provided with steel bosses which fit into the rubber buttons in the tray slats. The trays are made to contain any desired number of cells to suit different conditions in the service.

USEFUL INFORMATION

Specific Gravity: Specific Gravity is a combination of weight and volume. Liquids and solids are 1.

$$\text{Specific Gravity} = \frac{\text{Weight of Body in Air}}{\text{Weight of Water Displaced}}$$

The instrument to measure liquids lighter than water, is called a Baume Hydrometer.

Substance	Specific Gravity
Copper	8.8 to 9.0
Glass	2.4 to 2.8
Cast Iron	7.0 to 7.2
Lead	11.34
Steel	7.8
Zinc	6.9 to 6.2
Cork	.22 to .26
Pine (White)	.411
Oak (Red)	.69
H_2SO_4	18 (85%)
Oil, Mineral	90 to 94
Water, 4° Cent.	1.0
Sea Water	1.02 to 1.93

Care of Battery In Cold Weather:

A greenish deposit sometimes exists on the terminals of a storage battery which has been stored. This deposit may be removed with a solution of bicarbonate of soda (common cooking soda) in water. Do not allow any of this solution to get into the cells of the battery.

If the battery has not been kept charged during the winter, it is advisable to remove it from the line and have a plant equipped to take care of the work. Give it a fifty-hour charge at a 4-ampere rate, before putting it into service again.

The following is a table of the freezing temperatures of sulphuric acid and water solutions of specific gravities from 1.050 to 1.300:

Specific Gravity (Hydrometer Reading)	Freezing Temperature (Degrees Fahr.)
1.050	27 degrees
1.100	18 degrees
1.150	5 degrees
1.164	0 degrees
1.200	—17 degrees
1.250	—61 degrees
1.275 to 1.300	—90 degrees

Care should be taken when laying up the battery in cold weather. The battery should be fully charged and put away in a dry place.

WEIGHT IN POUNDS OF VARIOUS METALS

	Per Cu. Ft.	Per Cu. In.		Per Cu. Ft.	Per Cu. In.
Wrought Iron	480	.2778	Lead	711	.4114
Steel	490	.2836	Silver	655	.3790
Cast Iron	450	.2607	Gold (cast)	1204	.6968
Copper, Rolled	548	.3171	Platinum	1342	.7766
Brass, Rolled	524	.3032	Aluminum	159	.092

CHAPTER XVI.

RULES

GENERAL RULES AND REGULATIONS PRESCRIBED BY THE BOARD OF SUPERVISING INSPECTORS UNITED STATES STEAM BOAT INSPECTION SERVICE, DEPARTMENT OF COMMERCE, COVERING LAWS OF UNITED STATES MOTOR SHIPS.

Engineers of Motor Vessels

No person shall receive an original license as engineer or assistant engineer of motor vessels who has not served at least 36 months in the engineer's department of a motor vessel, a portion of which experience shall have been obtained within the three years next preceding the application:

Provided, That any person holding a license as engineer of steam vessels shall be eligible for license as engineer of motor vessels after having served for not less than three months as oiler in the engine department of motor vessels, or employed for not less than three months in the construction and installing of engines for motor vessels, which experience shall have been obtained within the three years next preceding the application; and any person who has served three years as apprentice to the machinist trade in a marine, stationary, or locomotive engine works, and any person who has served for a period of not less than three years as a locomotive or stationary engineer, and any person graduated as a mechanical engineer from a duly recognized school of technology may be licensed to serve as an engineer of motor vessels after having had not less than one year's experience in the engine department of motor vessels, a portion of which experience shall have been obtained within three years next preceding his application, which fact shall be verified by the certificate, in writing, of the licensed engineer or master under whom the applicant has served, said certificate to be filed with the application of the candidate.

No original license shall be granted any engineer or assistant engineer who can not read and write and does not understand the plain rules of arithmetic.

Inspectors may designate upon the certificate of any chief or assistant engineer the tonnage of the vessel upon which he may act. (Sec. 4426, R. S.)

Chief Engineers of Motor Vessels

An applicant for license as chief engineer of motor vessels shall be eligible for examination after he has furnished satisfactory documentary evidence to the local inspectors that he has had the following experience:

First: One year's service as first assistant engineer of motor vessels; or,

Second: Two years' service as second assistant engineer of motor vessels, or two years' combined service as first and second assistant engineer on motor vessels; or,

Third: One year's service as assistant engineer on motor vessels for license as chief engineer of motor vessels of 750 gross tons and under; or,

Fourth: Any person holding a license as chief engineer of steam vessels, after having served as oiler in the engine department of motor vessels for not less than three months or employed for not less than three months in the construction and installation of engines for motor vessels; or,

Fifth: Any person who has served at least one year in the engine department of motor or steam vessels, or who has had at least two years' experience in the construction of marine motor engines and their installation, shall be eligible for examination for license as chief engineer of motor vessels of not over 150 gross tons. (Sec. 4426, R. S.)

First Assistant Engineer of Motor Vessels

An applicant for license as first assistant engineer of motor vessels shall be eligible for examination after he has furnished satisfactory documentary evidence to the local inspectors that he has had the following experience:

First: One year's service as second assistant engineer of motor vessels; or,

Second: Two years' service as third assistant engineer of motor vessels, or two years' combined service as second and third assistant engineer of motor vessels; or,

Third: Three years' service as oiler in the engine department of motor vesels for license as first assistant engineer of motor vessels of 1,000 gross tons and under; or,

Fourth: Any person holding a license as first assistant engineer of steam vessels, after having served as oiler in the engine department of motor vessels for not less than three months or employed for not less than three months in the construction and installation of engines for motor vessels. (Sec. 4426, R. S.)

Second Assistant Engineer of Motor Vessels

An applicant for license as second assistant engineer of motor vessels shall be eligible for examination after he has furnished satisfactory documentary evidence to the local inspectors that he has had the following experience:

First: One year's service as third assistant engineer of motor vessels; or,

Second: Thirty-six months' actual service in the engine department of motor vessels, 12 months of which shall have been as oiler; or,

Third: Three years' service as an apprentice to the machinist trade and engaged in the construction or repair of marine, stationary, or locomotive engines, together with one year's service in the engine department of motor vessels as oiler; or,

Fourth: Any person holding a license as second engineer of steam vessels, after having served as oiler in the engine department of motor vessels for not less than three months or employed for not less than three months in the construction and installation of engines for motor vessels. (Sec. 4426, R. S.)

Third Assistant Engineer for Motor Vessels

An applicant for license as third assistant engineer of motor vessels shall be eligible for examination after he has furnished satisfactory documentary evidence to the local inspectors that he has had the following experience:

First: Two years' service as oiler on motor vessels; or,

Second: A graduate from the engineering class of a nautical school ship, the term of such engineering class to be based upon a period of two years, after he has served at least six months as oiler on motor vessels, or employed at least six months in the construction and installation of engines for motor vessels; or,

Third: A journeyman machinist who has been engaged in the construction or repair of marine motor engines for two years, together with one year's service in the engine department of motor vessels as oiler; or,

Fourth: Two years' service as a locomotive or stationary engineer, together with one year's service as oiler on motor vessels; or,

Fifth: A graduate in mechanical engineering from a duly recognized school of technology, together with six months' service as oiler on motor vessels; or,

Sixth: Any person who has completed the intensive training course prescribed by the United States Navy and who has been commissioned as ensign in the United States Naval Reserve Force may, upon the recommendation of the engineer officer or officers under whom he has served, be examined for license as third assistant engineer of motor vessels, after having actually served, after being commissioned, not less than 12 months as junior engineer officer on motor vessels; or,

Seventh: Any person holding a license as third assistant engineer of steam vessels, after having served as oiler in the engine department of motor vessels for not less than three months or employed for not less than three months in the construction and installation of engines for motor vessels. (Sec. 4426, R. S.)

RULE V.—LICENSED OFFICERS

ORIGINAL LICENSES

1. Before an original license is issued to any person to act as a master, mate, pilot, or engineer he shall personally appear before some local board or a supervising inspector for examination. Any person who has attained the age of 19 years and has had the necessary experience shall be eligible for examination: **Provided**, That no person shall receive a license as master, first mate, second mate, chief engineer, first assistant engineer, or second assistant engineer before reaching the age of 21 years.

Inspectors shall, before granting an original license to any person to act as an officer of a vessel, require the applicant to make written application upon the blank form furnished by the Department of Commerce, to be filed in the inspector's office. When practicable, applicants for master's, mate's, pilot's, or engineer's license shall present to the inspectors, to be filed with their application, discharges or letters from the master or other officer under whom they have served, certifying to the name of the vessel and in what capacity the applicant has served under him; also period of such service. Inspectors shall also, when practicable, require applicant for pilot's license to have the written indorsement of the master and engineer of the vessel upon which he has served, and of one licensed pilot, as to his qualifications. In the case of applicants for original engineer's license, they shall also, when practicable, have the indorsement of the master and engineer of a vessel on which they have served, together with one other licensed engineer.

The first license issued to any person by a United States inspector shall be considered an original license, where the United States records show no previous issue to such applicant.

No original license shall be issued to any naturalized citizen on less experience in any grade than would have been required of a citizen of the United States by birth.

On and after July 1, 1922, no candidate for original license as master, mate, pilot, or engineer shall be examined unless he shall present satisfactory evidence to the inspectors that he has completed a course of instructions in the principles of first aid approved by the United States Public Health Service for this particular purpose, and not until he presents a certificate from the United States Public Health Service, duly attested, that he has passed a satisfactory oral examination based upon the contents of the "Manual on Ship Sanitation and First Aid," or some other manual arranged for the purpose, having the approval of the United States Public Health Service. (Sec. 4405, R. S.)

VISUAL EXAMINATIONS REQUIRED FOR ORIGINAL AND RENEWED LICENSES

2. No original license as master, mate, or pilot of any vessel shall be issued except upon the official certificate of a surgeon of the Public Health Service respecting the vision of the person applying for such original license. The word "original," as contemplated in this section, shall mean the first license of any character issued to a master, mate, or pilot, and shall not be held to mean, for instance that a license issued to a master who was previously licensed as a mate or pilot shall be considered an original master's license.

No license as master, mate, or pilot of any class of vessel shall be renewed except upon the official certificate of a surgeon of the Public Health Service that the color sense of the applicant renewal is normal.

When an applicant for renewal of license is situated so that it would put him to great inconvenience or expense to appear before a surgeon of the Public Health Service for examination, the certificate of a reputable physician or oculist as to the color sense of the applicant shall be accepted in lieu of the certificate of the sugeon of the Public Health Service.

In case an applicant for original license or renewal of license is pronounced color-blind he may, in the discretion of the inspectors, be limited to act as master, mate, or pilot on a vessel navigating in daylight only.

Nothing herein contained shall debar an applicant who has lost the sight of one eye from securing a renewal of his license, providing that his color sense is normal. (Secs. 4439, 4440, 4442, R. S.)

EXAMINATIONS

3. No original master's, mate's, pilot's, or engineer's license shall be issued hereafter or grade increased except upon written examination by a board of local inspectors or a supervising inspector, which written examination shall be placed on file in the office of the inspectors issuing said license: Provided, however, That upon navigable waters where the only pilots obtainable are illiterate Indians or other natives, the fact that such persons can neither read nor write shall not be considered a bar to such Indians or other natives receiving license as pilot of steam vessels, providing they are otherwise qualified therefor.

Before granting or renewing a license inspectors shall satisfy themselves that the applicants can properly hear the bell and whistle signals.

When any person makes application for license it shall be the duty of the local inspectors to give the applicant the required examination ass soon as practicable. (Secs. 4405, 4439, 4440, 4441, 4442, R. S.)

REEXAMINATIONS AND REFUSAL OF LICENSES

4. Any applicant for license who has been duly examined and refused may come before the same local board for reexamination at any time thereafter that may be fixed by such board, but he shall not be examined by any other local board until one year has expired from the date of the refusal without the sanction of the board that refused the applicant.

If the inspectors shall decline to grant the applicant the license asked for, they shall furnish him a statement, in writing, setting forth the cause of their refusal to grant the same. (Secs. 4405, 4455, R. S)

PREPARATION OF LICENSES.

5. All licenses hereafter issued to masters, mates, pilots, and engineers shall be filled out on the face with pen and black ink instead of typewritten. Inspectors are directed, when licenses are completed, to draw a broad pen and black-ink mark through all unused spaces in the body thereof, so as to prevent, as far as possible, illegal interpolation after issue.

Every person receiving license or certificate of lost license shall sign same upon back thereof immediately upon its receipt. Sec. 4405, R. S.)

CERTIFICATE OF LOST LICENSE

6. In case of license of any class from any cause any board of local inspectors upon receiving satisfactory evidence of such loss and a record of the lost license from the board that issued same shall issue a certificate to the owner thereof, which shall have the authority of the lost license for the unexpired term, unless in the meantime the holder thereof shall have the grade of his license raised, after due examination, in which case a license in due form for such grade may be issued. In all cases where a certificate of lost license is issued by a board other than the board that issued the lost license the certificate of lost license shall state what board issued the lost license. (Sec. 4405, R. S.)

PARTING WITH LICENSE

7. Any license granted to a master, mate, pilot, engineer, or operator shall be immediately revoked if, for any purpose, the holder thereof voluntarily parts with its possession or places it beyond his personal control by pledging or depositing it with another. (Sec. 4405, R. S.)

RENEWAL OF LICENSE

8. Whenever an officer shall apply for a renewal of his license for the same grade, the presentation of the old license, with satisfactory certificate of visual examination, where required, and with oath of office, shall be considered sufficient evidence of his title to renewal, which old license and oath of office shall be retained by the inspectors upon their official files as the evidence upon which the license was renewed: Provided, That it is presented within 12 months after the date of its expiration, unless such title has been forfeited or facts shall have come to the knowledge of the inspectors which would render a renewal improper; nor shall any license be renewed more than 30 days in advance of the date of expiration thereof, unless there are extraordinary circumstances that shall justify a nenewal beforehand, in which case the reasons therefor must appear in detail upon the records of the inspectors renewing the license.

Whenever an officer shall apply for renewal of his license for same grade, after 12 months after the date of its expiration, he shall be required to pass an examination for the same grade of license. The renewed license in either case shall receive the next higher number for number of issue of present grade and for number of issues of all grades.

Whenever a licensed officer makes application for a renewal of his license, he shall appear in person before some board of local inspectors or supervising inspector, except that upon renewal of such license for the same grade, when the distance from any local board or supervising inspector is such as to put the person holding the same to great inconvenience and expense to appear in person, he may, upon taking oath of office before any person authorized to administer oaths, and forwarding the same, together with the license to be renewed and certificate of visual examination where required, to the local board or supervising inspector of the district in which he resides or is employed, have the same renewed by the said inspectors, if no valid reason to the contrary be known to them; and they shall attach such oath to the stub end of the license, which is to be retained on file in their office: *Provided, however,* That any officer holding a license, and who is engaged in a service which necessitates his continuous absence from the United States, may make application in writing for renewal and transmit the same to the board of local inspectors, with his certificate of citizenship, if naturalized, and a statement of the applicant, verified before a consul or other officer of the United States authorized to administer an oath, setting forth the reasons for not appearing in person; and upon receiving the same the board of local inspectors that originally issued such license shall renew the same and shall notify the applicant of such renewal, and no license as master, mate, or pilot of any class of vessel shall be renewed without furnishing a satisfactory certificate of color-blindness. (Secs. 4405, 4438, R. S.)

EXTENSION OF ROUTE AND RAISE OF GRADE OF LICENSES

9. Licensed officers serving under five years' license, entitled by license and service to raise of grade, after passing examination, shall have issued to them new licenses for the grade for which they are qualified, the local inspectors to file in their office the old license when surrendered, with the report of the circumstances of the case, but the grade of no license shall be raised except as hereinafter provided, unless the applicant can show one year's actual experience in the capacity for which he as been licensed.

Inspectors shall, before granting an extension of route or raise of grade of license, require the applicant to make his written application upon the blank form of application for extension of route or raise of grade of license furnished by the Department. When practicable, applicants for extension of route or raise of grade of license shall present to the inspectors, to be filed with the application, discharges or letters from the master or other officer under whom they have served, or other satisfactory documentary evidence, certifying to the name of the vessel and in what capacity the applicant has served; also period of such service.

If any board of local inspectors is satisfied by the documentary evidence submitted that a pilot is entitled by experience and knowledge to unlimited tonnage, it may remove any tonnage restrictions which may have been placed upon his license by any other board of local inspectors.

Except as hereinafter provided, practical service in the deck department of an ocean or coastwise vessel propelled by machinery shall be accepted when offered in documentary evidence by any person applying for an original license or raise of grade as equal to the same amount of service in any ocean or coastwise steam passenger vessel.

Service on United States lighthouse tenders propelled by machinery shall be considered as equivalent experience for raise of grade as that obtained on vessels subject to inspection by this Service.

Service on United States light vessels propelled by machinery shall be considered as one-half experience for raise of grade as that obtained on vessels subject to inspection by this Service. (Sec. 4405, R. S.)

EXAMINATION FOR RENEWAL OF MASTER'S OR PILOT'S LICENSE

10. It shall be the duty of all inspectors, before renewing an existing license to a master or pilot of steam vessels, for any waters, who has not been employed as master or pilot on such waters during the three years preceding the application for renewal to satisfy themselves, by an examination in writing, or orally, to be taken down in writing by the inspectors, that such officers are thoroughly familiar with the pilot rules upon the waters for which they are licensed. (Secs. 4439, 4442 R. S.)

LAWS, GENERAL RULES AND REGULATIONS, AND PILOT RULES TO BE FURNISHED LICENSED OFFICERS

11. Every master, mate, pilot, and engineer of vessels shall, when receiving an original license, a renewed license, or a raise of grade of license, be furnished by the inspectors with a copy of the Laws Governing the Steamboat-Inspection Service, and a copy of the General Rules and Regulations Prescribed by the Board of Supervising Inspectors, and every master and pilot of vessels and operator of motor vessels shall, when receiving an original license, a renewed license, or a raise of grade of license, be furnished by the inspectors with a pamphlet copy of the rules and regulations governing pilots and of the statutes upon which such rules are founded, applicable to the waters on which their licenses are intended to be used, as stated in the body thereof. (Sec. 4405, R. S.)

SUSPENSION AND REVOCATION OF LICENSES

12. When the license of any master, mate, pilot, or engineer is revoked such license expires with such revocation, and any license subsequently granted to such person shall be considered in the light of an original license except as to number of issue. And upon the revocation or suspension of the license of any such officer said license shall be surrendered to the local inspectors or supervising inspector ordering such suspension or revocation.

When the license of any master, mate, engineer, or pilot is suspended the inspectors making such suspension shall determine the term of its duration, except that such suspension shall not extend beyond the time for which the license was issued.

The suspension or revocation of a joint license shall debar the person holding the same from the exercise of any of the privileges therein granted, so long as such suspension or revocation shall remain in force. (Secs. 4450, R. S.)

13. Whenever a supervising, local, or assistant inspector of steam vessels, or any of them, shall find on board any vessel subject to the provisions of Title LII of the Revised Statutes any licensed officer under the influence of liquor or other stimulent to such an extent as to unfit him for duty, or when any licensed officer shall use abusive or insulting language to any inspector or assaults any such inspector while on official duty, the local inspectors or the supervising inspector shall immediately suspend or revoke the license of the officer so offending without further trial or investigation.

The fact of a licensed officer being under the influence of liquor in the presence of the inspector or inspectors to such an extent as to unfit him for duty while on board a vessel shall be sufficient cause for such suspension or revocation. (Secs. 4405, 4450, R. S.

LICENSES TO OFFICERS OF VESSELS OWNED BY THE UNITED STATES

14. Any person who has served at least one year as master, commander, pilot, or engineer of any steam vessel owned and operated by the United States in any service in which a license as master, mate, pilot, or engineer was not required at the time of such service shall be entitled to license as master, mate, pilot, or engineer, if the inspectors upon written examination, as required for applicants for original license, may find him qualified: *Provided*, That the experience of any such applicant within three years of making application has been such as to qualify him to serve in the capacity for which he makes application to be licensed. (Secs. 4439, 4440, 4441, 4442, R. S.)

EXTRACTS FROM RULES OF AMERICAN BUREAU OF SHIPPING

SECTION 36, INTERNAL COMBUSTION ENGINES

CONTENTS

GENERAL CONDITIONS FOR CLASSIFICATION

(1) Request for Survey

The construction of main and auxiliary internal combustion engines intended for Classification with the American Bureau of Shipping is to be carried out in accordance with the following requirements, under the supervision and to the satisfaction of the Surveyors.

Before proceeding with the manufacture of materials subject to test and inspection (as listed in Paragraphs 5 to 12) builders are required to notify the Bureau in writing that survey is desired.

(2) Drawings and Data to be Submitted

Builders are to submit blue prints in triplicate of the following drawings for the main engines: Bedplate or crankcase, engine cylinder including jacket and liner, shafting, connecting and piston rods, sectional assemblies of engine parts, and the air containers. For auxiliary oil engines, prints of the following drawings are to be submitted: General outline, crankshaft, connecting rod and engine cylinder.

The Committee reserves the right to require the submittal of such additional drawings as in its judgment are necessary to determine the safety of the installation. Particulars of the engines and sufficient data for calculating the stresses are to be supplied together with the drawings.

The Committee is prepared to examine and comment on additional drawings upon the request of the Owners or Builders.

(3) Supplementary Requirements

The requirements for boilers, pumps, piping, electrical equipment, steering gear, etc., as specified in other Sections of the Rules apply also to vessels fitted with Internal Combustion Engines, unless otherwise specified in this Section.

(4) Classification

Upon satisfactory completion in accordance with the requirements of the Rules, the machinery will be entered in the Record Book ✠ A. M. S. signifying the Highest Classification of the American Bureau of Shipping for Machinery and Boilers, and Special Survey during construction.

The Committee reserves the right to refuse class to any vessel where the engines have not been built under the Survey of the Bureau.

MATERIAL REQUIRING TEST AND INSPECTION

(5) Modification for Test and Inspection

The material listed below is to be made in accordance with the requirements of the Rules and to be tested and inspected by the Surveyors to the Bureau. Copies of purchase orders for material requiring test and inspection at the source, stating the requirements of the purchaser, should be forwarded for the information of the Surveyors.

(6) Forgings and Castings

The material for the following is to be in accordance with Section 40, Para. 8 and Para. 11.

For engines of all sizes: Crankshafts, thrustshafts, lineshafts, propeller shafts, also generator shaft and motor shafts for indirect drive.

For engines with cylinders 12 inches in diameter and over: connecting rods, piston rods and frame tension rods.

For engines with cylinders 18 inches in diameter and over: cross-

head pin, shaft coupling bolts, connecting rod bolts and main bearing bolts.

(7) Brass and Copper Tubes

Seamless copper and brass tubes for compressed air intercoolers and after coolers and copper tubes for injection air and starting air are to be in accordance with Section 40. Paras. 18 and 19 respectfully.

(8) Seamless Steel Pipes

Seamless steel pipes for high pressure fuel oil and for injection air and starting air are to be in accordance with Section 40, Para. 14.

(9) Lap-Welded Steel Pipe

Lap-welded steel pipes for starting air and for starting air containers carrying pressures not exceeding 350 lbs. are to be in accordance with Section 40, Para. 13.

(10) Seamless Containers

Seamless steel containers for injection air and for starting air carrying pressures exceeding 350 lbs. are to be in accordance with Section 40, Para. 15.

(11) Riveted or Welded Containers

The material for riveted or welded containers is to be in accordance with section 40, Para. 1 to 5.

(12) Test of Other Materials When Requested

The Surveyors to the Bureau will test other materials than those listed above upon request from the builders or owners.

GENERAL INSTALLATION REQUIREMENTS

(13) Arrangement of Machinery

Drawings showing the general arrangement of the machinery in the vessel giving sizes and types of the various engine auxiliaries and ship auxiliaries, also the sizes and purposes of the suction and discharge connections of the pumps, are to be submitted for approval.

(14) Ventilation

All parts of the engine room shall be thoroughly ventilated by efficient supply and exhaust vents.

(15) Floor and Gratings

It is recommended that the engine room platforms be made of open grating. Supports are to be of metal.

(16) Sheathing

Steel Hulls should have all wood work eliminated from the machinery space.

In wood and composite vessels, planking, frame timbers, bulkheads, etc., in the machinery compartment are to be metal sheathed from the floors six feet up, and all woodwork within four feet of unjacketed cylinders, exhaust pipes and silencers and within six feet directly over the cylinders of all oil engines having flame starting torches, is to be metal sheathed and lagged with asbestos mill board not less than one-half inch thick.

(17) Oil Drain Well

The overflows of fuel oil, the drains from fuel and lubricating tanks and from drip pans of oil pumps and oil tanks are to be led to a closed cofferdam or drain tank fitted with air and sounding pipes and having an independent suction to the fuel oil transfer pump. Special care should be taken to prevent the drainage of fuel and lubricating oil into the bilges.

Where a gutter bar is fitted to intercept the seepage from deep oil tanks the well thus formed should be drained into the oil drain tank or cofferdam by way of drain valves, or have a suction to the fuel transfer pump.

(18) Fire Extinguishers

The means provided for extinguishing fire must be thorough and effective. Hose connections from the fire main with sufficient hose to cover every part of the engine room, are to be located in accessible positions within easy reach and always available for service.

Portable fire extinguishers of approved manufacture and of about 2½ gal. capacity each should be stored as directed in the machinery space. The number of extinguishers to be provided is as follows:

On steel vessels: 2 on vessels with propelling engines up to 300 B.H.P. and 1 additional extinguisher for each additional 300 B.H.P.

On wooden vessels: 3 extinguishers for propelling engines up to 200 B.H.P. and 1 additional extinguisher for each additional 200 B.H.P.

The maximum number of extinguishers required for single screw vessels is 8 and for twin screw vessels 10. It is recommended that on vessels with large power a permanent chemical fire extinguishing system of approved type be installed with hose connections in approved locations in the engine room; in this case the number of portable extinguishers need not exceed three.

(19) Engine Room Electrical Apparatus

The electrical installation is to be in accordance with the requirements of Section 37.

ENGINE CONSTRUCTION

(20) General

The following requirements apply to all oil engines for propelling and auxiliary purposes. All machinery parts subject to stresses are to be

of sound material and the fits and clearances in accordance with the best marine practice. The passages for cooling water and lubricating oil must be carefully cleaned of sand and scale. The main bearings and the reciprocating parts should be readily accessible and lifting eyes or gear are to be fitted in way of main bearings and cylinder covers. The nuts of main bearing and connecting rod bolts are to be secured by split pins or other efficient means.

Hand or power turning gear shall be provided for all oil engines. The engines for propelling the vessel are to be fitted with a governor or other efficient means to prevent the speeding of the engines to more than 15 per cent above the designed number of revolutions; propelling engines over 300 B.H.P. should be direct reversible. Closed crankcase engines should have suitable provisions to prevent the accumulation of gas in the crankcase.

(21) Bedplate

The bedplate or crankcase is to be of rigid construction, oil tight, and to be provided with a sufficient number of bolts to secure the same to the ships structure. The structural arrangement for supporting and securing the main engines are to be sumitted for approval.

(22) Cylinders

Cylinders, liners, cylinder covers, pistons and other castings subject to high temperatures or pressures are to be made of the best grade of cast iron or equally satisfactory material. Castings must be free from defects affecting their strength.

Cylinders using a compression pressure of over 400 lbs. per sq. inch are to fitted with relief valves set to not more than 1 1-3 times the maximum working pressure, and the valve discharge should lead beyond a point of danger to life or vessel.

(23) Crankshafts

The minimum diameter of the cranshaft is to be determined by the following formula:

$$d = a \sqrt[3]{\frac{D^2PL}{f}}$$

d=Diameter of shaft in inches
D=Diameter of cylinder in inches
P=Initial Working Pressure in lbs. per sq. in.
L=Fore and aft length of crank over webs plus 1 (inch)
a=Factor from table below
S=Stroke of piston in inches
t=Thickness of crankweb in line with axis of shaft
w=Width of crankweb perpendicular to axis of shaft
f=7500 for Grade 1 Forgings, Section 40-11.
f=8000 for Grade 2 Forgings, Section 40-11.
f=6500 for cast steel made in accordance with Sec. 40, Par. 8

The value of P given by the builders must be verified by the Surveyor from indicator cards during the full power trial of the engine. A set of full power indicator cards of the main engines is to accompany the Classification Report.

Subsequent adjustments for the purpose of obtaining higher initial pressures are not permitted without the special approval of the Committee.

VALUES OF "a" FOR AIR INJECTION DIESEL ENGINES

Number of Cylinders		S/L Ratios							
4 cycle	2 cycle	.7	.8	.9	1.0	1.1	1.2	1.3	1.4
1-2-4	1-2	1.17	1.19	1.22	1.25	1.28	1.31	1.34	1.36
3-6	3	1.19	1.22	1.25	1.28	1.32	1.35	1.38	1.41
8	4	1.20	1.24	1.27	1.30	1.33	1.37	1.40	1.43
12	6	1.22	1.25	1.29	1.32	1.36	1.39	1.42	1.45
16	8	1.25	1.29	1.33	1.36	1.40	1.44	1.47	1.50

VALUES OF "a" FOR EXPLOSIVE COMBUSTION ENGINES

Number of Cylinders		S/L Ratios							
4 cycle	2 cycle	.7	.8	.9	1.0	1.1	1.2	1.3	1.4
1-2-4	1-2	1.17	1.17	1.17	1.17	1.17	1.17	1.17	1.17
3-5-6	3	1.17	1.17	1.17	1.17	1.19	1.20	1.22	1.24
8	4	1.17	1.19	1.21	1.23	1.25	1.28	1.30	1.32
10-12	5-6	1.18	1.20	1.23	1.25	1.28	1.31	1.33	1.35

NOTE: The above constants are to be used in cases where a bearing adjoins each side of each crank and where single impulses occur at equal intervals. In cases of departure from these conditions and for designs not properly coming under the two groups above, data must be submitted for the determination of shaft dimensions.

The dimensions of the crankwebs of solid shafts are to be such that wt^2 is not less than $.4d^3$ and w^2t not less than d^3; for built-up crankshafts t is to be not less than .66d and w not less than 1.9 diameters of the hole in the webs. These porportioned dimensions are based on the use of the same grade of material for both shaft and webs and should be modified in accordance with the difference in the grade of the materials.

The webs are to be shrunk or forced unto the shaft and are to be fitted with dowels or keys of ample proportions to transmit at least 60% of the torque. It is strongly recommended that the radius of the fillets in way of main bearings and the crank pins of solid shafts be not less than .05d. The shearing stress in the coupling bolts should not exceed the torsional stress in the shafting.

(24) Intermediate Shafts

The diameter of the intermediate shafts is to be determined by the following formula:

$$d=b\sqrt[3]{\frac{D^2PS}{f}}$$

VALUES OF "b"

Air Injection Diesel Engines Number of Cylinders			Explosive Combustion Engines Number of Cylinders		
Four Cycle	Two Cycle	b	Four Cycle	Two Cycle	b
1-2-3-4-6	1-2-3	.97	1-2-3-4	1-2	.87
8	4	1.00	5-6	3	.94
12	6	1.04	8	4	1.00
16	8	1.10	10-12	5-6	1.05
	12	1.20		8	1.10

Shafting for Auxiliary or indirect drive engines may be 5% less in diameter than required by the above formulae.

(25) Thrustshafts

To be 5 per cent larger in diameter than the intermediate shafts.

(26) Propeller Shafts

To be the diameter of the thrust shafts multiplied by the value of C in the table for Propeller Shafts in Section 34, Paragraph 3. The ratio in this case being that of the diameter of the Propeller to the diameter of the **thrust** shaft.

(27) Trial

Before final acceptance all engines must demonstrate by a running test their ability to perform satisfactorily the work for which they are intended.

(28) Recommendation

In order to minimize operating troubles it is recommended that a system of periodical inspection, cleaning, replacement and adjustments of essential engine parts be followed by the ship's engineers.

ENGINE AUXILIARIES

(29) General

The following auxiliaries are intended to cover the minimum requirements for seagoing vessels of average power and should be increased for ships of large power but may be modified for River and Harbor Boats and for sailing vessels fitted with auxiliary engines. Propelling engines of a type and design not requiring certain auxiliaries noted below are to be provided with the necessary auxiliaries or connections in duplicate. (See also para. 13.)

Pressure gauges, thermometers and relief valves are to be fitted where required.

(30) Fuel Oil Transfer Pumps

One attached pump for each engine and one independent pump, or two independent pumps.

(31) Compressors for Injection Air

SINGLE SCREW: One attached or one independent compressor of maximum capacity, and one idependent compressor of 66% capacity.

TWIN SCREW: (a) One attached or one independent compressor per engine of maximum capacity, and one independent compressor of 66% capacity for one engine.

(b) One independent compressor of maximum capacity for both engines and one compressor of 66% capacity for both engines.

(c) One attached compressor for each engine, capable and arranged to supply both engines for maximum requirements and one starting air compressor of adequate size.

NOTE: The capacities noted above refer to the maximum requirements at any speed.

(32) Emergency Air Compressors

One power operated compressor not requiring air for starting; capacity depending on size of propelling units.

(33) Scavenging Pumps (for two cycle engines)

One or more attached pumps for each engine or one independent pump. Crankpit scavenging will be approved for explosive combustion engines.

(34) Main Lubricating Oil Pumps

One attached pump for each engine and one independent pump, or two independent pumps; each of a capacity for maximum requirements. Not required in cases of general multi-feed lubrication.

(35) Water Cooling Pumps

One attached pump for each engine and one independent pump, or two independent pumps; each of a capacity for maximum requirements.

(36) Note

Independent pumps denote pumps driven independently of the propelling engines.

Attached pumps denote pumps driven by the propelling engines.

ENGINE PIPING

(37) Fuel Oil Transfer System

The piping arrangements for the carriage of fuel oil are to be in accordance with the requirements of Section 31.

The service tanks are to be located sufficiently high to permit gravity flow to the service pump suctions and shall be fitted with air vents leading to the atmosphere, drain cocks and a well protected oil

gauge with valves on both ends. Tanks are to be placed in drip pans provided with a drain to the oil drain well.

The pumps used for the transfer of oil are not to be used for bilge and ballast purposes; the suction pipe is to be fitted with an efficient strainer.

(38) Fuel Oil Injection System

The suction to the fuel oil injection pumps is to be fitted with a Duplex Strainer and cut-out valves are to be located at the service tanks operated from the engine room floor. The piping in the discharge line should be of seamless drawn steel and the fittings of extra heavy steel; means should be provided to discontinue the fuel supply to each individual cylinder. The joints in the pipe lines should be metal to metal or have metal gaskets; ample provision is to be made for expansion.

Individual injection pumps for each cylinder are recommended.

Overflows and drip pans are to have drain pipes leading to the oil drain well.

(39) Starting Arrangement

The efficiency and the capacity of the starting arrangement as required in Para. 49 is to be demonstrated to the Surveyor in attendance.

On vessels fitted with Surface Ignition Engines all reasonable safeguards must be provided to prevent fires. (See also Par. 16.)

(40) Scavenging System

The air supply is to be adequate for all speeds and the flow of the air should be regulated to insure uniform supply. The suction and discharge valves of scavenging pumps shall be readily accessible for examination and repair. Provision should be made to take care of excessive pressure by means of relief valves or breaking plates.

(41) Injection Air System

The discharge lines of each stage of the air compressors are to be fitted with air coolers and relief valves of ample proportions. The temperature of the air discharge from each cooler should not exceed 150°.

A stop valve is to be fitted in the branch line to each cylinder so located that the engine may be operated with one or more cylinders cut off. The pipe lines may be made either of seamless steel or seamless copper and should be fitted with drains.

The installation of oil separators is strongly recommended.

(42) Lubricating Oil System

The lubrication of main bearings, crankpins, crosshead or wrist pins should be by means of pump pressure or by gravity. The pistons of the engines, compressors and scavenging pumps should be lubricated by individual pumps, adjustable to the particular requirements.

Extreme care must be taken to prevent the contamination of the oil supply by sea water, escaping oil gases or carbon. The oil drawn from the crank pit or sump must run through a duplex strainer, readily accessible for cleaning. The installation of oil filters or separators is recommended.

The tanks for the storage of lubricating oil must not form part of the ship's structure.

The flashpoint of the lubricating oil should in no case be below 350°F. In order to insure the continued safe operation of the engines, only the best quality of lubricants should be used.

(43) Cooling Water System

The cooling water pumps are to have at least one high and one low sea suction; the suction line is to be fitted with duplex strainers. For any emergency connection from the fire main the cooling water should be led through the duplex strainer. Means should be provided to ascertain the temperature of the return from each cylinder and the proper circulation through all water jackets.

Drain cocks must be provided at the lowest point of all jackets and the discharge must be led to the bilge. A relief valve should be fitted in the main line to the jackets to prevent excessive pressure due to leaks.

(44) Exhaust Piping

The exhaust pipes should be water jacketed or efficiently insulated. Where a number of cylinders are connected to one exhaust pipe, allowance should be made for expansion. Exhaust pipes of several engines should not be connected but run separately overboard, unless such connection can be made as will prevent the return of gases to an idle engine. Boiler uptakes and the engine exhaust lines should not be connected.

AIR CONTAINERS

(45) Thickness of Material

The thickness in inches of the material in wrought steel containers is to be determined by the following formulae:

(46) Shell $$t = \frac{2PD}{SE}$$

For seamless tubes $E = 1$

For lap-welded tubes $E = .75$

For riveted plates $E = \dfrac{p - d}{p}$

For the rivets $E = \dfrac{.85na}{pt} \times \dfrac{s}{S}$

When rivets are in double shear, the value of E for the rivets is to be multiplied by 1.875. The lower efficiency of the two formulae for riveted points is to be used in the formula for the thickness of the shell.

The efficiency of the circumferential end lap joints is not to be less than 60% of the efficiency of the longitudinal point.

(47) Convex and Concave Heads

For convex heads $t = \dfrac{2.5PR}{S}$

For concave heads, and for convex heads fitted with a manhole, the thickness found by the above formula is to be multiplied by 1.2. The thickness of the heads shall in no case be less than the thickness of the shell.

(48) Flat Heads $t = .9 \sqrt{\dfrac{D^2P}{S}}$

Unless properly reinforced by ribs, cast steel heads shall be 20 per cent thicker than required by the above formula.

t=Thickness in inches

P=Working Pressure in lbs. per sq. in.

D=Greatest inside diameter of shell in inches

s=Tensile Strength of rivet bar

S=Min. tensile strength of plate in lbs. per sq. in.

E=Efficiency of joint

p=Pitch of rivets in inches

d=Diameter of rivet holes in inches

n=Number of rivets per pitch

a=Area of rivet hole

R=Major radius of dished head, maximum 1.5D

Unless otherwise specified on the approved plan S equals the minimum tensile strength allowed by the Rules, Section 40, Para. 3.

(49) Volume of Starting Air

It is recommended that each vessel be provided with not less than two containers. The container capacity for directly reversing engines is to be sufficient for at least twelve consecutive starts and for non-reversing engines for 6 consecutive starts for each engine without recharging. The Surveyor is to verify this capacity by a trial of the engine.

Each container or group of containers is to be fitted with a cut-out and a relief valve or its equivalent.

(50) Containers for Spray Air

Engines using spray air are to be provided with at least two containers per vessel, so arranged that each may be used independently of the other.

It is strongly recommended that containers be fitted with copper breaking plates. Safety valves shall be fitted between containers and compressor delivery and in the absence of the container breaking plates, also to the container.

(51) General Requirements

The number of connections on all containers should be reduced to a minimum. The containers are to be so installed as to make the drain connections effective under extreme conditions of trim. Connections should be provided for cleaning tanks and pipe lines. For material requirements see Pars. 9 to 11.

SHIP AUXILIARIES

(52) Donkey Boilers

Where a donkey boiler is located in the engine room the same is to be fitted with guard plates and drip pans in the way of the furnaces. Boilers installed for the purpose of providing power for ship and engine auxiliaries are to have at least two means of feeding and two oil fuel service pumps. The Rules in Section 32 apply for the construction of Donkey Boilers.

(53) Bilge and General Service Pumps

Bilge and General Service Pumps are to be provided as specified for steam vessels in Section 31. At least two pumps, one of which must be independent of the main engine shall have discharge connections to the fire main.

(54) Electric Generating Sets

At least two generators for lighting the vessel and for operating the electric auxiliaries are to be fitted, each of a capacity for full requirements at sea. Where three units are provided, each should be of a capacity of at least 50 per cent of the full requirements at sea. Generators should be placed in a location where damage by oil or water is remote and be fitted with guard plates and handrail.

TESTS

(55) Hydrostatic Tests

The following hydrostatic tests are to be witnessed and recorded by the Surveyor. Castings, fittings and pipes showing defects during or subsequent to these tests are to be rejected. All relief and safety valves are to be tested and set in the presence of the Surveyor.

*Engine Cylinders or liners to a distance of 20 per cent of the working stroke.	1 2/3 times the working pressure
Engine Cylinder covers in water jacket.	100 lbs. per sq. in.
All other water jackets.	50 lbs. per sq. in.
Cylinders of air compressors and their coolers and pipes.	1 2/3 times the working pressure.
Scavenging pump cylinder and piping.	15 lbs. per sq. in.
Lubricating oil system.	50 lbs. per sq. in.
Injection air bottles.	1 2/3 times the working pressure.
Starting air containers.	1 2/3 times the working pressure.
Fuel oil transfer system.	50 lbs. per sq. in.
Steam lines.	2 times working pressure.
Feed lines.	2½ times the working pressure.
Sump tank.	15 lbs. per sq. in.
Fuel oil service tanks.	15 lbs. per sq. in.

*Where Cylinders or liners are so designed that the parts subject to internal pressure may be accurately gauged for thickness of material and such thickness is not less than one twelfth of the diameter of the cylinder, the hydrostatic test may be dispensed with.

(56)

The piping systems for starting air, fuel injection and injection air are to be tight under their relief valve pressure.

SPARE PARTS AND EQUIPMENT

(57) For Single and Twin Screw Installations

*1 Main engine cylinder head complete with valves, cages, springs, etc.

1 Exhaust valve per engine complete with cage, spring, etc.

1 Intake Air Valve complete with cage, spring, etc.

1 Fuel Valve complete with cage, spring, etc.

25% of fuel valve needles or the equivalent.

1 Starting air valve complete with cage, spring, etc.

1 Cylinder relief valve complete with cage, spring, etc.

1 Set of piston rings for one piston.

1 Fuel pump complete or the working parts for one cylinder.

1 Piston lubricating pump complete or the working parts for one cylinder.

1 Complete set of air compressor piston rings for each size piston.

10% of the valves for all air compressors at least one of each size and type, complete with cages and seats.

25% of the plungers for multifeed lubricators.

2 connecting rod bolts and nuts—top end.

2 connecting rod bolts and nuts—bottom end.

2 Main bearing bolts and nuts of each size.

¼ Set of coupling bolts for one coupling of each size.

1 Spring of each size and type fitted.

25% of each size of gaskets and packing, at least one of each kind.

*5% of engine and compressor cylinder head studs.

One set of the above of each size and design, for main and auxiliary engines are to be supplied.

At least one valve of each type and size for oil transfer pumps, lubricating oil pumps, cooling water pumps and scavenging pumps.

*A sufficient length of each size of pipe used for injection air, starting air and injection oil lines to replace the longest pipe.

*Assorted bolts, nuts, pipe flanges and pipe couplings.

A set of templets and gauges for adjusting gear and aligning main bearings.

*A sufficient amount of bearing metal to rebabbitt the largest bearing.

A book of instructions for operating, maintaining and overhauling the main and auxiliary engines.

Items marked * not required for river and harbor boats.

SURVEYS

(58) Annual Survey

Machinery installations with Internal Combustion engines are subject to annual Survey, at which a general examination is to be made of the main and auxiliary engines. The Surveyers should be given the opportunity to examine such machinery parts as may be opened for inspection or repair.

The main engine crank shaft is to be checked for alignment.

The main and auxiliary oil engines shall be given a running test in the presence of the Surveyor and the maximum working pressure must not exceed the pressure for which the engines are approved.

At least one cylinder of each engine, and all air compressor cylinders are to be opened up and the interior of the cylinder, the piston and piston rings are to be examined; the air, fuel and safety valves are to be in-

spected. If all the above are found satisfactory, the cylinder thus examined may be taken as representing the general condition of the engine; if any part is not satisfactory, similar parts of the other cylinders are subject to examination at the discretion of the Surveyor.

Where the engineers of the vessel are required to make systematic periodical examinations and replacements of the essential machinery parts, as recommended in Paragraph 28 and the report of such examinations and replacements is entered in the engineers log, the inspection of the engine parts enumerated above may be omitted at the discretion of the Surveyor.

One section each of the injection air and starting air lines are to be removed and if found oily, the air lines, air containers and air coolers, are to be cleaned.

The engine room bilges are to be inspected and the causes of any oil leakage into the same to be remedied.

The fire extinguishing apparatus shall be recharged when required, to the satisfaction of the Surveyor.

The spares must be checked in accordance with the requirements of Para. 57.

(59) Special Periodical Surveys

The requirements for Special Surveys as specified in Section 45, apply also to main and auxiliary internal combustion engines as far as applicable.

The various engine piping systems, the air containers, coolers, oil tanks and the engine auxiliaries are to be thoroughly cleaned and retested in accordance with Paragraph 55. The cylinders of all oil engines, air compressors and scavenging pumps shall be opened up for examination.

The crank shaft shall be lifted and the lower bearing shells be examined and rebabbitted where required. Other parts of the machinery as may be considered necessary by the Surveyor are to be opened for examinations. The spares must be checked in accordance with the requirements of Para. 57.

(60) Survey of Machinery Not Built Under American Bureau Survey

The machinery is to be surveyed, inspected and tested as required under Special Periodical Survey. The general workmanship, the condition of the machinery and where possible the physical characteristics of the shafting shall be reported by the Surveyor.

The allowable working pressure will be determined upon the submission of the required engine data and sizes of shafting.

The requirements with regard to fire protection must be complied with in every case. The whole machinery installation shall be brought up to the requirements of the Rules or the equivalent to the satisfaction of the Surveyor.

LLOYD'S RULES FOR THE CONSTRUCTION AND SURVEY OF DIESEL ENGINES AND THEIR AUXILIARIES

SECTION 1. In vessels propelled by Diesel Oil Engines, the Rules as regards machinery will be the same as those relating to steam engines, so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes, and propellers.

Construction

SECTION 2. In vessels built under Special Survey and fitted with Diesel Engines, the engines must also be constructed under Special Survey.

2. In cases of Diesel Engines being built under Special Survey, the distinguishing mark ✠ will be noted in Red, thus: ✠LMC or ✠NE.

3. In order to facilitate inspection, the plans of the machinery are to be examined by the Surveyors, and the dimensions of the shafts are to be submitted for approval.

4. The Surveyors are to examine the materials and workmanship from the commencement of the work until the final test of the machinery under full working conditions; any defects are to be pointed out as early as possible.

5. Any novelty in the construction of the machinery is to be reported to the Committee and submitted for approval.

6. The auxiliary engines used for air compressing, working dynamos and ballast, or other, pumps, are also to be surveyed during construction.

7. In cases where the designed maximum pressure in the cylinders does not exceed 500 lbs. per square inch, the diameters of the crank shaft of the main engines are not to be less than those given by the following formula:

$$\left.\begin{array}{l}\text{Diameter of}\\ \text{crank shaft}\end{array}\right\} = \sqrt[3]{\mathrm{D}^2 \times (\mathrm{AS}+\mathrm{BL})}$$

where D = diameter of cylinder,

S = length of stroke,

L = span of bearings adjacent to crank, measured from inner edge to inner edge.

The value of $(\mathrm{AS} + \mathrm{BL})$ are as given in the following table:

Table I

4-Cycle Single Acting Engine	2-Cyle Single Acting Engine	Values of the Co-efficients
4 or 6 cyls.	2 or 3 cyls.	$.089S + .056L$
8 cyls.	4 cyls.	$.099S + .054L$
10 or 12 cyls.	5 or 6 cyls.	$.111S + .052L$
16 cyls.	8 cyls.	$.131S + .050L$

For auxiliary engines of the Diesel Type the diameters of the crank-shafts may be five per cent. less than given by the foregoing formula.

8. In solid forged shafts the breadth of the webs should be not less than 1.33 times and the thickness not less than 0.56 times the diameter of the shaft as found above, or, if these proportions are departed from, the webs must be of equivalent strength.

9. The diameter of the intermediate shaft must not be less than that given by theformula:

$$\left.\begin{array}{l}\text{Diameter of inter-}\\ \text{mediate shaft} \ldots\ldots\end{array}\right\} = C \sqrt[3]{D^2 \times S}$$

where D = the diameter of cylinder,

S = the stroke of piston,

C is a co-efficient found from the following table by interpolation from the values found for A.

Where the stroke is not less than 1.2 times, nor more than 1.6 times the diameter of the cylinder, $(.735\ D + .273\ S)$

may be taken instead of $\sqrt[3]{D^2 \times S}$.

Table II

2-Cyle Single Acting Engines	Values of the Co-efficient C where		
	$A = .0025$	$A = .0050$	$A = .0100$
2 Cyls.	.305	.317	.336
3 Cyls.	.346	.363	.385
4 Cyls.	.364	.380	.396
5 Cyls.	.380	.391	.404
6 Cyls.	.398	.403	.412

4-Cycle Single Acting Engines	Values of the Co-efficient **C** where A = .0025	A = .0050	A = .0100
4 Cyls.	.300	.312	.327
6 Cyls.	.338	.355	.370
8 Cyls.	.357	.366	.376
10 Cyls.	.376	.382	.389
12 Cyls.	.394	.398	.404

In using the above table the appropriate value of **A** is found from

$$A \times W \times d^2 \times R^2 = D^2 \times S$$

where **D** = diameter of cylinder in inches,

S = stroke of piston in inches,

d = diameter of flywheel in feet,

R = revolutions of engines per minute,

W = total weight of flywheel in tons.

10. The diameter of the flywheel shaft must be at least equal to that of the crank shaft.

11. Where ordinary deep collars are used the diameter of the thrust shaft measured under the collars must be at least 21/20ths that of the intermediate shaft. The diameter may be tapered off at each end to the same size as that of the intermediate shaft.

12. The diameter of the screw shaft must be not less than the diameter of the intermediate shaft (found as above) multiplied by $\left(.63 + \frac{.03P}{T}\right)$ but in no case must it be less than 1.07**T**,

where **P** = the diameter of the propeller in inches,

T = the diameter of intermediate shaft in inches.

The size of the screw shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube, as provided for in Section 11, paragraph 3, of the Rules for Engines and Boilers for Steam Vessels. If no liners are used, or if two separate liners are used, the diameter of the screw shaft should be 21/20ths that given above.

The diameter of the screw shaft is to be tapered off at the forward end to the size of the thrust shaft.

13. If the designed maximum pressure in the cylinders exceeds 500 lbs. per square inch, the diameters of the shafting throughout must be increased in the proportion of $\sqrt[3]{\frac{\text{Maxim. press. in lbs. per sq. in.}}{500}}$

14. Where the cylinder liners are made of hard close grained cast iron of plain cylindrical form, accurately turned on the outside as well as bored on the inside so that their soundness can be ascertained by inspection, and their thickness at the upper part is not less than 1/15th of the diameter of the cylinder, they need not be hydraulically tested by internal pressure. If, however, they are made of complicated form, the question of testing must be submitted.

15. The water jackets of the cylinders, and the water passages of the cylinder covers and pistons, must be tested by hydraulic pressure to 30 lbs. per square inch, and must be perfectly tight at that pressure.

16. The exhaust pipes and silencers must be water-cooled or lagged by non-conducting material, where risk of damage by heat is likely to occur.

17. The cylinders are to be fitted with safety valves loaded to not more than 40 per cent. above the designed maximum pressure in the cylinders and discharging where no damage can occur.

18. The air compressors and their coolers are to be so made as to be easy of access for overhaul and adjustment.

19. Where the fuel is injected into the cylinders by air pressure, the following conditions are to observed:

In single screw vessels, an auxiliary air compressor is to be provided of sufficient power to enable the main engines to be kept continuously at work when the main compressor is out of action.

If the manoeuvering gear is arranged so that the engines can be kept continuously at work with some of the cylinders out of action, the auxiliary compressor need only be of sufficient power to enable the engines to be kept at work under these conditions.

In twin screw vessels in which two sets of compressors are fitted, the auxiliary compressor must be of such size as to enable it to take the place of either of the main compressors. If in such engines each main compressor is sufficiently large to supply both engines, a smaller auxiliary compressor will be sufficient.

20. A small auxiliary compressor, worked by a steam engine, or by an oil engine not requiring compressed air, is to be fitted for first charging the air receivers.

21. At least one high pressure air receiver is to be arranged with connections to enable it to be used for fuel injection, in case the working receiver of either main engine is out of use from any cause.

22. The circulating pump sea suction is to be provided with an efficient strainer which can be cleared inside the vessel.

23. In all vessels fitted with engines in which the lubricating oil is circulated under pressure a spare oil pump is to be supplied with all connections ready for immediate use, and two independent means are to be arranged for circulating water through the oil cooler.

AIR RECEIVERS AND PIPES

SECTION 3. 1. Compressed air receivers for starting air are to be supplied of sufficient capacity to permit of twelve consecutive startings of the engines without replenishment.

2. Cylindrical receivers for containing air under high pressure, used either for starting or for the injection of fuel in oil engines, may be made either of seamless steel or of welded, or riveted, steel plates.

3 **Quality of Metal.**—If made of welded, or riveted, steel plates, the ordinary rules regarding steel material for boilers apply, which provide that where welding is employed, either in the longitudinal seams or at the ends, the material must have a tensile strength not exceeding 30 tons per square inch (Section 33, par. 7, Rules for Engines and Boilers). In these cases the welding must be lap welding; neither oxy-acetylene nor electric welding will be permitted.

4. In the case of seamless receivers, the rules for material will be the same as for boiler shells, but the permissible extension may be 2 per cent less than that required with boiler plates.

5. **Tensile and Bend Tests** are to be made from the material of **each** receiver. When they are welded or riveted, the tests may be made, and the thicknesses verified before the plates are bent into cylindrical form. In the cases of seamless receivers, the thicknesses must be verified by the Surveyor before the ends are closed in, and at this time the Surveyor shall select and mark the test pieces required from either of the open ends of the tube. The test pieces are to be annealed before test, so as to properly represent the finished material.

6. The permissible working pressure for welded or seamless receivers is to be determined by the following formula:
Maximum working pressure in lbs. per square inch

$$= \frac{C \times S \times (T - 2)}{D}$$

for thicknesses of 5/8 in. and above,

$$= \frac{C \times S \times (T - 1)}{D}$$

for thicknesses below 5/8 in.,

where **S** = Minimum tensile strength of the steel material used, in

T = Thickness of the material, in sixteenths of an inch.

D = Internal diameter of cylinder, in inches,

C = Co-efficient as per following table:

Co-efficient—

77 for seamless receivers of thickness of 5/8 in. and above,

69 for seamless receivers of thickness below 5/8 in.

54 for welded receivers of thickness of 5/8 in. and above.

48 for welded receivers of thickness below 5/8 in.

7. for flat ends welded into the cylindrical shells, the thickness must not be less than

$$T = \frac{D}{17} \times \sqrt[3]{P}$$

where T = thickness, in sixteenths of an inch,

D = internal diameter, in inches,

P = working pressure, in lbs. per square inch.

8. The permissible working pressure for receivers made of riveted steel plates is to be determined by the rules regulating the working pressure of boilers.

9. Each welded or seamless receiver shall be carefully annealed after manufacture, and before the hydraulic test.

10. Each welded or seamless receiver shall be subjected to a hydraulic test of twice the working pressure, which it shall withstand without permanent set.

11. Each receiver made of riveted steel plates for pressures up to 300 lbs. per square inch is to be tested by hydraulic pressure 1½ times the working pressure, plus 50 lbs. per square inch. Where higher working pressures are used, the test pressure need not be more than 200 lbs. per square inch above the working pressure.

12. All receivers above six inches internal diameter must be so made that the internal surfaces may be examined, and, wherever practicable, the openings for this purpose should be sufficiently large for access. Means must be provided for cleaning the inner surfaces by steam, or otherwise.

13. Each receiver which can be isolated must have a safety valve fitted, adjusted to the maximum working pressure. If, however, the air compressor is fitted with a safety valve so arranged and adjusted that no greater pressure than that permitted can be admitted to the receivers, they need not be fitted with safety valves.

14. Each receiver must be fitted with a drain arrangement at its lowest part, permitting oil and condensed water to be blown out.

15. Oil or air pipes subjected to high pressure are to comply with the Rules for steam pipes, Section 13, Clauses 7 and 16 (Rules for Engines and Boilers of Steam Vessels).

Pipes which are subjected to a working pressure up to 1,000 lbs. per square inch must be tested hydraulically to at least twice the working

pressure to which they will be subjected, and those subjected to a higher working pressure than 1,000 lbs. per square inch to an hydraulic test of at least 1,000 lbs. per square inch above their working pressure.

PUMPING ARRANGEMENTS

SECTION 4. The pumping arrangements are to be the same as would be required for steam vessels of similar size and power, with the exception that no bilge suction need be fitted to the main engine circulating pump. In the cases of vessels fitted with water ballast, the water ballast pump must have, in addition, one direct suction from the engine room bilges.

GENERAL

SECTION 5. 1. All oil fuel pipes, tanks and their fittings must comply with the requirements of Section 49 (Rules for Steel Ships).

2. Special attention must be given to the ventilation of the engine room.

3. If the auxiliaries are worked by electricity, the cables in connection with them must be in accordance with the rules for electric fittings.

SPARE GEAR

SECTION 6. The articles mentioned in the following list will be required to be carried, viz.:

1 cylinder cover complete for the main engines, with all valves, valve seats, springs, etc., fitted to it.

In addition, one complete set of valves, valve seats, springs, etc., for one cylinder of the main and of the auxiliary Diesel engines, and fuel needle valves for half the number of cylinders of each engine.

1 piston complete, with all piston rings, studs, and nuts for the main engines.

In addition, one set of piston rings for one piston of the main and of the auxiliary Diesel engines.

1 complete set of main skew wheels for one main engine.

2 connecting rod, or piston rod, top-end bolts and nuts, both for the main and for the auxiliary Diesel engines.

2 connecting rod bottom end bolts and nuts, both for the main and for the auxiliary Diesel engines.

2 main bearing bolts and nuts, both for the main and for the auxiliary Diesel engines.

1 set of coupling bolts for the crank shaft.

1 set of coupling bolts for the intermediate shaft.

1 complete set of piston rings for each piston of the main and of the auxiliary compressors.

1 half set of valves for the main and for the auxiliary compressors.

1 fuel pump complete for the main engine, or a complete set of all the working parts.

1 fuel pump for the auxiliary Diesel engine, or a complete set of all working parts.

1 set of valves for the daily fuel supply pump.

1 set of valves for the water circulating pumps.

1 set of valves for one bilge pump.

1 set of valves for the scavenge pump, where lift valves are used.

1 set of valves for the lubricating oil pump.

1 bucket and rod for the lubricating oil pump.

A quantity of assorted bolts and nuts, including one set of cylinder cover studs and nuts.

Lengths of pipes suitable for the fuel delivery and the blast pipes to the cylinders, and the air delivery from the compressors to the receivers, with unions and flanges suitable for each.

PERIODICAL SURVEYS

SECTION 7. 1. The engines are to be submitted to survey annually, and in addition are to be submitted to a Special Survey upon the occasion of the vessels undergoing the Special Periodical Surveys Nos. 1, 2, and 3 prescribed in the Rules, unless the machinery has been specially surveyed within a period of twelve months, in which case the Annual Survey will be sufficient. The boilers, if fitted, are to be subjected to the same surveys as required by Section 37 of the Rules for Engines and Boilers of Steam Vessels.

2. **Special Surveys.**—At these special surveys, the main engines and the auxiliary engines are to be examined throughout, viz.:—All the cylinders, pistons, valves and valve gears, connecting rods and guides, pumps, crank, intermediate, and thrust shafts, propellers, stern bushes, sea connections and their fastenings, are to be examined. The air compressors are also to be examined. The air receivers are to be cleaned and examined and, if necessary, tested, as provided for in paragraph 3 of this Section.

3. **Annual Surveys.**—The whole of the parts of the engines which the engineers of the vessel open up for adjustment and overhaul should be examined and reported upon. The Survey must include, for each main engine, the examination of at least 2 pistons, 2 cylinder covers and their valves, 2 connecting rods and their brasses, both top and bottom ends. 2 of the main bearings and crank shaft journals, and 1 of the tunnel bear-

ings. If these are all satisfactory, their condition may be taken as representing that of the other similar parts.

In the auxiliary Diesel engines, a similar course must be adopted, but in this case one of each of the parts mentioned of each engine will be sufficient, if found to be satisfactory.

The valve gears of all the Diesel engines should be examined, as far as practicable, without complete dismantling.

The air receivers must be examined internally if possible, and, together with the air pipes from the compressors, must be cleaned internally by means of steam, or otherwise. If the air receivers cannot be examined internally, they must be tested by hydraulic pressure to twice the working pressure at each alternate Annual Survey, attention being specially given to the welding of the ends and of the longitudinal joints.

The pumps and air compressors must be examined and tried under working conditions. If found to be satisfactory ,they need not be dismantled.

The manoeuvering of the engines must be tested under working conditions.

If the examination reveals any defects, the Surveyor should recommend such further opening up as he may consider to be necessary.

4. **Record of Survey.**—If the various parts mentioned in paragraphs 2 or 3 are all found to be in a satisfactory condition and the Surveyor finds that the machinery generally is in good order, he should recommend the vessel to have a fresh record of LMC.

5. **Survey of Screw Shafts.**—The screw shaft is to be examined annually and drawn at intervals as provided for in Section 37, Clause 3 (Rules for Engines and Boilers of Steam Vessels).

PRECAUTIONS AGAINST DANGER

The time has long passed, when the use of oil on shipboard is opposed on account of insurmountable danger. Oil has the distinct advantage that it is not subject to spontaneous combustion, and many fires which have occured in ships' bunkers at sea would not have been possible with oil. Certain precautions, however, must be taken—such as suitable arrangements of vent pipes, protection of bunker bulkheads, if exposed to heat, and particularly the use of oil with a reasonable high flash point.

The United States Navy, in cooperation with the Bureau of Mines, has investigated this matter of possible explosions of gases in storage tanks, and it was found that no inflammable gases were formed in any amount in the storage tanks or bunkers until the oil was heated to the flash point, i.e., that the representative oil tested contained no dissolved gas or vapor sufficient to form an explosive mixture at temperatures below the flash point. The largest percentage of vapor in the atmosphere of fuel tanks of various ships tested, was 0.04%, whereas, about 0.9% is required to form an explosive mixture. It was also found that any oil in the bunker tank had to be heated to within 60 degrees F. of the flash point before even a faint "glow" of partial burning was obtained on introducing a naked flame in the tank.

These important investigations show that oil is perfectly safe on board ship, so long as the flash point is sufficiently above the temperature to which the oil may be exposed.

On the other hand, while careful attention to ventilation of the tanks and leading the vent pipes well away from all possible chance of exposure to flame, may result in immunity from trouble. The conclusion is forced upon us that the use of heavy oils which have to be heated in the tanks and bunkers may lead to very serious consequences through the necessity of installing heating coils in the tanks, and the possibility that the oil becoming heated to the flash point through carelessness. This of course should be understood, is rarely the case in regards to Fuel used in Diesel engines.

Where torches are in use, as on Semi-Diesel engines, necessitating the heating of the hot bulbs. etc., if the simple precaution is taken of always having the lighted torch under the burner before turning on the oil, no possible danger of explosion in the engine room can exist.

Several methods of extinguishing fires at sea by the use of carbonic acid gas are being developed, such as the Gronwald system, advanced by leading Fire Syndicates, which consist of the installation of tanks at suitable points containing liquid carbonic acid gas under high pressure. These tanks are piped to various parts of the ship, where possible danger from fire might exist, and the gas is admitted to these points in emergency, thus, completely blanketing the fire and shutting off the supply of oxygen. Another system which has been very effective in extinguishing fires in oil tanks, is that known as the Erwin system, manufactured by

the Treadwell & Company of New York. A mixture of bicarbonic of soda and soap bark is carried in one tank, and sulphuric acid is carried in another, nearby, and these may be mixed automatically or at will, resulting in the liberation of a large mass of foam impregnated with carbonic acid gas. Carbon tetrachloride has been used for extinguishing fires; this is the best known in commercial form in the tanks of Pyrene. It occurs to the layman, that quite as much danger may result from the installation of tanks of this highly asphyxiating material on board ship as would be caused by fire, but undoubtedly experience will show the efficiency as well as the necessity of these various methods of extinguishing fires.

INDEX OF ILLUSTRATIONS, DIAGRAMS AND VIEWS

GENERAL INDEX